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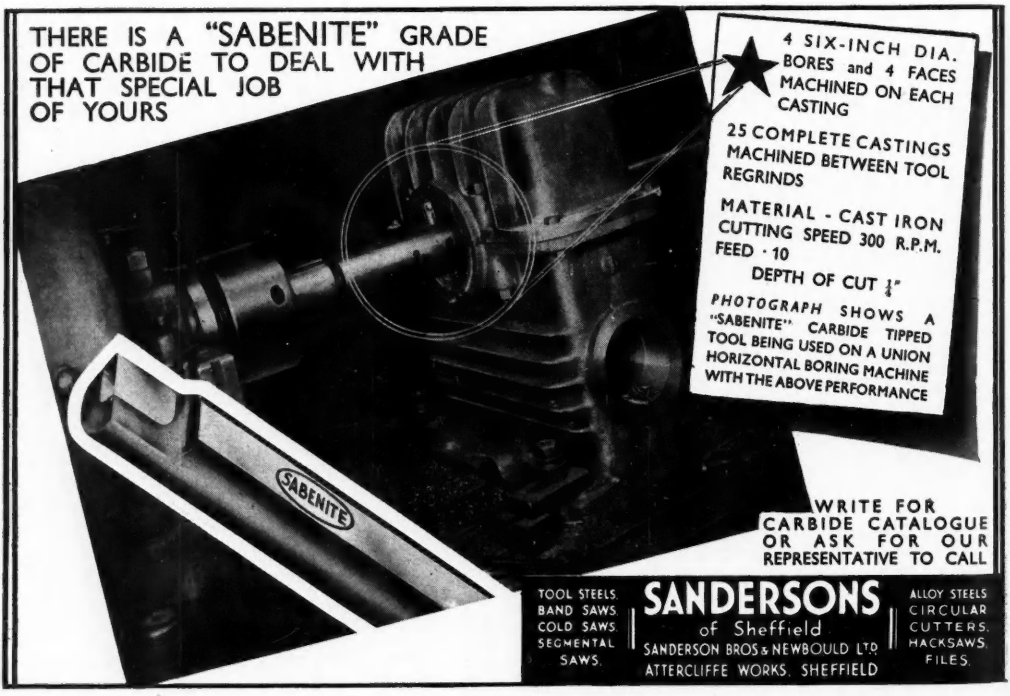
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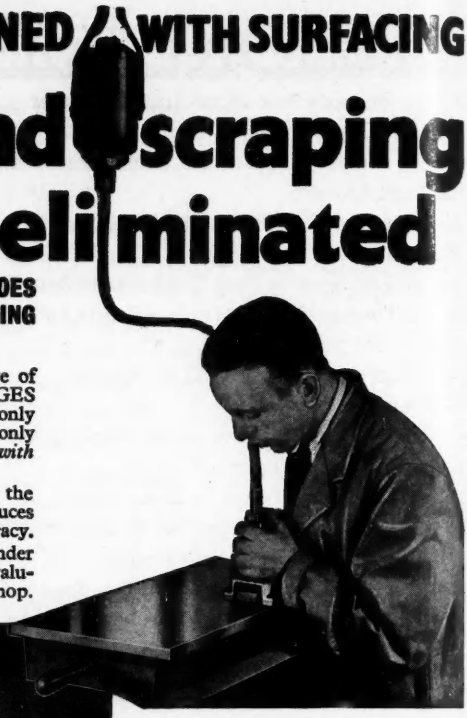

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IN THIS ISSUE

ALTHOUGH first attracting wide interest during the recent years of fuel shortage, the heat pump is not a new invention. The idea of using an engine for transforming quantities of low temperature heat to a higher temperature level was advanced by Lord Kelvin as early as 1852. Some years later his purely theoretical observations led to the development of the refrigerating machine, which, in fact, can be regarded as a special type of heat pump. The practical application of the heat pump as a heating machine, however, was not introduced until many years later, between the World Wars, when both Switzerland and the United States made considerable progress in its development. Whereas in Switzerland the dominating motive was fuel economy, in the United States the development of the heat pump had gained impetus by the possibility of combining it with air conditioning; moreover, it offered a new and very considerable outlet for the use of electricity. It is for this reason, perhaps, that Americans concentrate on the development of the heat pump for domestic purposes; Swiss engineers, on the other hand, are more conservative about the economics of this device for domestic uses and pay greater attention to its application for large-scale purposes, such as the heating of large public buildings, and industrial purposes.

In England, comparatively little attention has been paid to these developments, and apart from small experimental plants, e.g., T.G.N. Haldane's, the Norwich Heat Pump is believed to be the first and only large heat pump used for space heating in Britain. Expert opinion appears to be divided in this country, as can be seen from the discussion of J. A. Sumner's paper on the Norwich Heat Pump, read before a meeting of the Institution of Mechanical Engineers last year. However, it cannot be denied that this problem is of topical importance today. The article on page 405, although perhaps not containing revelations which are new to the expert, approaches the problem of the heat pump from the point of view of practical employment and attempts to define the circumstances under which the heat pump is of economic value, not only in times of shortage, but also under normal conditions. We hope, therefore, that this article will promote interest among a wide circle of engineers.

* * *

When silver brazing by induction heating is used for the joining of metal components, unit costs can often be cut by properly designing the parts to be joined. The article on page 419 gives many data, the importance of which is often underestimated by the designer. It is explained why the positioning of parts, the clearance between the surfaces, and the position of the silver alloy relative to the heating coils are important factors in producing strong and reliable joints; but the greatest value of this article lies in its instructive illustrations, the underlying ideas of which can easily be adopted in practice.

* * *

The article by Tcheglaev, see page 401, on the control of turbine power plants, describes various controlling systems and develops a simple method for the calculation of control processes using a two-stage amplification which gives greater sensitivity. The stability and response of the various systems are also compared.

* * *

The electrical industry has long recognised the difficulty of the problem of making satisfactory copper-aluminium joints. One solution to this problem, where outdoor cable connections are concerned, was recently announced in this journal.* Quite a different and apparently new approach to this problem is the development of upset welding, which is described on page 422 in this issue. This development has been brought about by the necessity of attaching a copper end to an aluminium coil for inter-coil connections, and making connection to coil terminals in electrical applications. Mechanical joints may in time build up an oxide film and develop high electrical resistance, with consequent heating, which might progress to the point of failure of the mechanical joint because of arcing and melting.

The use of solder has not been practical, since a soldered joint on aluminium can fail because of electrolytic corrosion resulting under certain conditions of high humidity and in contaminated atmospheres. Moreover, a soldered joint has little strength at the temperatures at which some of these joints might be required to operate.

* * *

Theoretical discussions regarding the best working conditions for wire-drawing are often considered useless in practice, especially when the results obtained are not supported by experimental evidence. Although this is not expressly confirmed in the report on page 398, readers are likely to come to the conclusion that the results given agree to a large extent with practical experience, and that the developments made and data presented should be of practical significance to all those concerned with the art of wire-drawing.

We extend to all readers
our best wishes
for a Happy Xmas
and a
Prosperous New Year.

*See THE ENGINEERS' DIGEST, September, 1948, page 309.

The Shape of Wire-Drawing Dies in Relation to the Drawing Effort and Friction Conditions

By E. Siebel, N. Ludwig, and P. Melchior. (From *Werkstatt und Betrieb*, Vol. 81, No. 7, July, 1948, pp. 177-180, 9 illustrations).

Drawing dies should be so shaped that the wire cross-section is reduced with minimum pull and minimum deformation work. The minimum theoretical work done in deforming a wire by drawing can be determined, with sufficient accuracy, from

$$A_N = V\,\varphi_h\,k_{fm} \quad .. \quad .. \quad (1)$$

where V is the deformed volume,

$$\varphi_h = \log_e \frac{F_0}{F_1} = \frac{q}{1-\frac{1}{2}q}$$

the degree of deformation i.e., the logarithm of the ratio of the original to the deformed cross-section, and q the area reduction. k_{fm}, the mean resistance to deformation of the wire, is taken as the arithmetic mean of the yield strength of the wire before and after the drawing pass. The work done in overcoming friction is, with sufficient accuracy for the purpose of this investigation, as follows:—

$$A_R = A_N \frac{\mu}{\alpha} = V\varphi_h\,k_{fm}\,\frac{\mu}{\alpha} \quad .. \quad (2)$$

where μ is the coefficient of friction and α half the entrant die angle. The work lost by inner shear of the material on entering and leaving the die, in the case of circular cross-sections[1], is:—

$$A_s = \frac{2}{3}\,A_N\,\frac{\alpha}{\varphi_h} = V\varphi_h\,k_{fm}\,\frac{2\alpha}{3\varphi_h} \quad (3)$$

The shear due to deformation within the die of non-planar cross-sections is negligibly small under normal conditions. The efficiency of the deformation process can, therefore, be determined from

$$\eta_F = \frac{A_N}{A_N + A_R + A_S} = \frac{1}{1+\frac{\mu}{\alpha}+\frac{2\alpha}{3\varphi_n}} \quad .. \quad (4)$$

Since the work lost in friction increases, whilst the work lost by inner shear decreases with an increasing entrant angle 2α, the efficiency must reach a maximum for a certain value of α. Fig. 1 shows that this occurs at an angle of inclination α (half the entrant angle) of 3 to 12 degrees, the higher values corresponding to heavier area reductions.

The mean tensile stress in the outgoing wire

$$\sigma_{1m} = k_{fm}\,\varphi_h/\eta_F \quad .. \quad .. \quad (5)$$

must be less than the tensile strength σ_t of the drawn wire, and a stress factor $a_F = \sigma_{1m}/\sigma_t$ can be formulated which must not exceed 75 per cent, taking into acccount additional stresses in the initial stage of a drawing pass. The maximum permissible deformation

$$\varphi_{h\max} = \sigma_{1m}\,\eta_F/k_{fm} = a_F\,\eta_F\,\sigma_t/k_{fm} \quad .. \quad (6)$$

therefore, cannot exceed 50 to 70 per cent, corresponding to an area reduction q of 40 to 50 per cent. In practice, an area reduction of 40 per cent is not exceeded in order to prevent cracks forming inside the material.

Knowledge of the coefficient of friction is the basis for calculation of deformation efficiency, stress factor and power requirements during drawing. Equation (4), can be rearranged, i.e.,

$$\mu = \left(\frac{1}{\eta_F} - 1 - \frac{2\alpha}{3\varphi_h}\right)\alpha \quad .. \quad (7)$$

which equation permits of comparing the qualities of different lubricants if the drawing tension is measured and the efficiency of deformation calculated. Tests[2] on the drawing of steel wire through tungsten carbide dies at a speed of 5 fps, and area reduction of 35 per cent, using lubricating soap, showed, for 6 S.W.G.

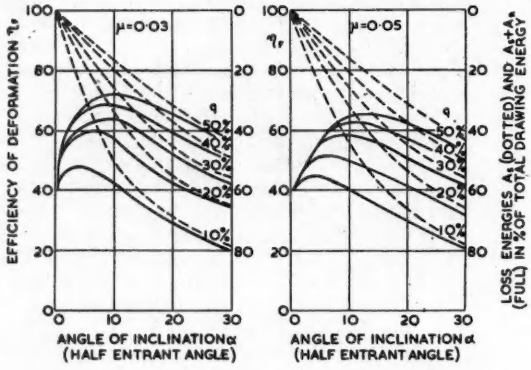


Fig. 1. **Efficiency of deformation as a function of nozzle form and relative reduction of cross-sectional area q per pass.** (μ = coefficient of friction).

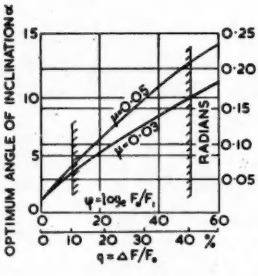


Fig. 2. (left) **Optimum angle of inclination α (half-entrant angle) during drawing.**

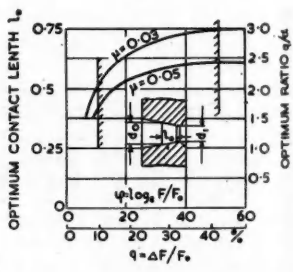


Fig. 3. (right) **Optimum length of contact between wire and die nozzle.**

annealed mild steel wire (0·06 per cent C), a coefficient of friction $\mu = 0{\cdot}03$ to $0{\cdot}04$, whereas for patented steel wire (0·53 per cent C), 8 S.W.G., μ was 0·01 to 0·015 and even dropped below 0·01 during the third pass. With vegetable oils friction was again 0·01 to 0·015 during the first and second passes of patented steel wire, but increased with the third pass. This increase was still more unfavourable when drawing oil and sodium soap were used. With these lubricants, the mild steel wire showed a friction of $\mu = 0{\cdot}04$ to 0·06. The surface of the patented steel wire is, thus, considerably more favourable than that of the annealed mild steel wire. In this connection, a brilliant polish of the drawing hole and an adherent lubricating film are extremely important. Adhesion can be improved by an incipient layer of rust obtained by exposure to air of the pickled wire after rinsing in water. Phosphate deposits which have proved favourable for the deep-drawing of sheet can be applied to wire-drawing. Thin deposits of ductile metals also reduce friction, e.g., copper plating, in a pickling bath, of wire subjected to the wet drawing process. Lead deposits have proved best for the drawing of stainless steels and highly alloyed high-tensile steels.

The entrant angles for optimum efficiency of deformation are shown in Fig. 2 for area reductions of 10 to 40 per cent as used in practice. For optimum efficiency, the angle should be adapted to the amount of area reduction and the friction to be expected, but, in general, it would be sufficient to choose a mean angle of inclination α (half-entrant angle) of 4 to 6 degrees for 10 to 20 per cent and 6 to 8 degrees for 20 to 40 per cent reduction of area. The lower values should be used for patented steel wire with its low coefficient of friction. The ratio of contact length between wire and die to the out-going diameter

$$\frac{l_1}{d_1} = \frac{d_0 - d_1}{d_1} \cdot \frac{1}{2\alpha} \cong \frac{\varphi_h}{4\alpha} \qquad \ldots \quad (8)$$

should be kept within certain limits[3]. Its optimum value is shown in Fig. 3 to be 0·5 to 0·7 under normal conditions. For large area reductions and low friction, it may be increased to 1·0.

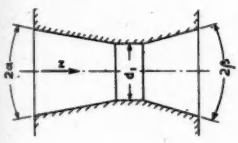


Fig. 4. **Drawing hole with straight taper.**

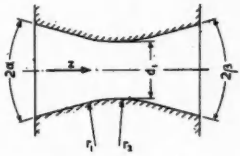
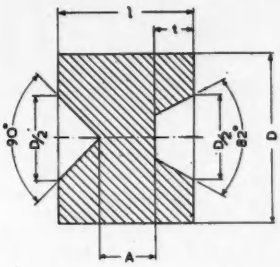


Fig. 5. **Drawing hole with variable inclination.**

The drawing hole can be drilled with a straight taper (Fig. 4) or variable inclination (Fig. 5). The straight taper is easier to manufacture and check. The angle between the incoming wire and die wall is fixed, which is an advantage for the formation of the oil film. With correct dimensioning of the cylindrical part, the outgoing wire diameter can be kept to close tolerances. On the other hand, optimum conditions are only obtained if the entrant angle is adapted to the amount of area reduction and an entrant angle below 12 degrees (angle of inclination 6 degrees) cannot insure perfect entry conditions for the wire. The straight taper should, therefore, only be used for area reductions above 20 per cent. The variable-inclination die (Fig. 5) insures optimum efficiency for any degree of area reduction under fixed friction conditions and offers advantages of tool standardization.



Sintered or pressed, dimensions in mm			
D	l	A	t
8	4	2	0
9	6	2·5	1·5
9	7·5	3	2·5
13	7	3	1

a

Sintered, dimensions in mm			
D	l	A	t
15	10	4	2·5
17	13	5	3·5
20	14	5·5	3·5
22	17	7	4·5
30	24	9·5	7

b

Fig. 6. **Normally used press forms for drawing dies.**

The moment seems opportune for a general standardization of drawing dies, taking into account the optimum calculated entrant angles as well as the widely employed die press forms (Fig. 6). Fig. 7 shows a proposal for the dimensioning of standard straight-taper dies with three different entrant angles graded according to the amount of area reduction. The cylindrical guide length z has been chosen rather short, so that the hole can be re-polished a number of times without unduly lengthening the cylindrical part.[4] The hole diameter d_1 (see Table) should be chosen with due regard to the established wire standards, but should not exceed a certain value, since the active part of the die hole must be restricted to, say, the middle third of the die length in order to obviate the danger of bursting. According to this condition, the incoming wire diameter is:

$$d_0 = d_{1\,\max} \sqrt{\frac{1}{1-q}} = d_{1\max} + 2\tan\alpha \left(l_1 - \frac{l}{3}\right) \qquad (9)$$

From eq. (9), with $l_1 = 0{\cdot}6\,l$ (Fig. 7), we obtain

$$d_{1\max} = \frac{8\,l\tan\alpha}{15\left[\sqrt{\frac{1}{1-q}} - 1\right]} = \frac{8\tan\alpha}{15\,C_1}\,l \qquad \ldots \quad (10)$$

Numerical evaluation shows $d_{1\max}$ to vary between $0{\cdot}475\,l$ (for $\alpha = 6$ degrees and $q = 20$ per cent) and $0{\cdot}324\,l$ (for $\alpha = 10$ degrees and $q = 40$ per cent). Taking $d_{1\max} = l/3$ to be a safe standard value, the allowable area reduction, from eq. (10) becomes

$$q = 1 - \frac{1}{\left(1 + \frac{8\tan\alpha}{5}\right)^2} \qquad \ldots \quad \ldots \quad (11)$$

The proposed standard drawing nozzle can, therefore, be used without danger of bursting, with a 12-degree entrant angle for area reductions from 20 to about 27 per cent, with a 16-degree entrant angle up to about 33 per cent, and with a 20-degree angle up to about 40 per cent reduction.

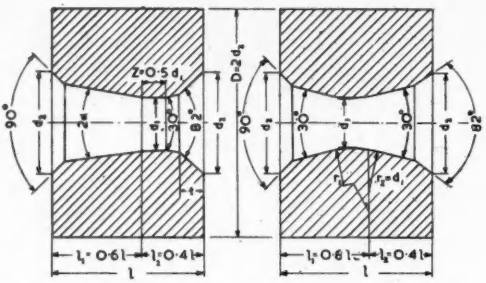


Fig. 7. **Proposed standard shape of die nozzle with straight entrant taper.**

Nozzle shape I_1 : $2\alpha = 12°$.
I_2 : $2\alpha = 16°$.
I_3 : $2\alpha = 20°$.

For area reductions between 20 and 40 per cent.

Fig. 8. **Proposed standard shape of die nozzle with decreasing entrant taper.**

Nozzle form II_1 : $r_1 = 4\,d_1$.
II_2 : $r_1 = 2{\cdot}5\,d_1$.
II_3 : $r_1 = 1{\cdot}5\,d_1$.

d_1[1])	D	l	l_1	l_2	t
0·5 to 1·3	4	4	2·5	1·5	0
1·4 „ 2·5	4·5	7·5	4·5	3	2·5
2·6 „ 4·25	8·5	13	8	5	3·5
4·5 „ 6·0	11	17	10	7	4·5
6·3 „ 8·0	15	24	14·5	9·5	7
8·5 „ 10·0	20	30	18	12	8

For drawing dies with steel mounting to German Standard DIN 1547-1944. Dimensions in mm.

d_1[1])	D	l	l_1	l_2	t
0·5 to 1·3	4	4	2·5	1·5	0
1·4 „ 2·0	9·5	6	3·5	2·5	1·5
2·1 „ 3·4	7·5	10	6	4	2·5
3·6 „ 4·75	10	14	8·5	5·5	3·5

For drawing dies with brass mounting to German Standard DIN 1547-1944. Dimensions in mm.

1) The diameters d_1 should be graded to German Standard DIN 2076.

Fig. 8 shows the proposal for standard variable-inclination die nozzles. According to the respective choice of radius r_1 which determines the inclination, the length of contact between wire and die wall becomes, for $q = 40$ per cent, $0{\cdot}7\,d_1$, $0{\cdot}9\,d_1$, or $1{\cdot}1\,d_1$, and for $q = 20$ per cent, $0{\cdot}4\,d_1$, $0{\cdot}5\,d_1$ or $0{\cdot}65\,d_1$. Even for small reductions, this length thus keeps within practical limits. The proposed standards make it possible to attain optimum conditions, as shown in Figs. 2 and 3, for any case of normal drawing practice. Dies for wires above 10 mm ($\frac{3}{8}$ inch) diameter have not yet been standardized, but the same considerations would apply. Concurrently with a standardization of the nozzle form, the press form standards (Fig. 6) should be reconsidered with a view to a more systematic grading of the diameters.

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Oxidation and Decarburization in Reheating and Rolling

By G. WALLQUIST. (From *Jernkontorets Annaler, Sweden*, Vol. 132, No. 6, June, 1948, pp. 175-212, 20 illustrations.)

THE technical research work of Jernkontoret has included studies regarding oxidation and decarburization of steel in reheating and rolling. To this end, not only have laboratory experiments been carried out under conditions similar to those existing in reheating furnaces, but practical service tests have also been made in some rolling mills. The present article deals with the theory of oxidation and decarburization, gives a short survey of previous research and describes the investigations made and their results.

As to the theory of oxidation and decarburization, a summary is given of equilibrium reactions and equilibrium diagrams presented by different authors. Earlier investigations have been directed toward determining the factors which cause a change in the steel surface. These factors are, above all, the steel composition, the gas composition, the gas velocity, the reheating temperature and the reheating time.

The laboratory experiments were made with twelve steels, comprising seven carbon steels, three low-alloy steels and two high-alloy stainless steels. The test specimens, stream-lined and 10 mm in diameter, were turned, ground and polished. The specimens were exposed to oxidation and decarburization in a stream of gas containing 20 per cent of a varying mixture of carbon monoxide and carbon dioxide and 80 per cent of nitrogen. The CO : CO_2 ratio was varied from 6 to 0, i.e., pure carbon dioxide, keeping the amount of nitrogen constant. In addition, the gas velocity, reheating temperature and re-heating time were varied.

The results of the laboratory experiments show the influence of the different factors on oxidation and decarburization. At the same time they confirm previous experience in that the amount of oxidation as well as decarburization is dependent above all, on reheating temperature and reheating time. Changes of gas composition and gas velocity which are possible in practical service can only affect the reheating result to a limited extent.

The service tests with two carbon steels containing 0·74 and 0·82 per cent carbon were made in five steel works. The decarburization was studied when rolling ingots to billets, 75 × 75 mm, and also when rolling these billets to 22 mm round bars. Parallel tests were made in different works. Some of the billets were rolled without surface treatment, and others after being pickled and ground clean. Reheating and rolling were done by the procedure normally adopted in every plant. For special studies of the variation of decarburization during rolling, test specimens were obtained after different passes during rolling as well as from the finished bars.

The results of the service tests show that, although the reheating and rolling conditions were somewhat different, the average decarburization was of the same magnitude in the different works. When ground billets are used, the decarburization is considerably smaller than in untreated billets, and this is especially evident with regard to the ferrite-rich zone of decarburization.

An irregular and varying depth of decarburization is evidenced in different billets and bars as well as in different surface elements of the same billet or bar. The amount of decarburization is charged during rolling, and it may increase or decrease at the same point of the surface. In certain parts, the surface may be quite free from decarburization, whereas a real concentration of decarburization may occur in other parts. This irregularity is due to variations in the flow of material at the surface in relation to the center of the bar, and this flow is most accentuated at high reductions.

Turbine Regulation Systems with Two Stages of Amplification

By A. V. TCHEGLAEV. (From *Izvestia Vsesoyuznogo Teplotechnicheskogo Instituta*, No. 8, August, 1947, pp. 1-6, 10 illustrations.)

WITH increased turbine powers it is becoming common practice to employ control systems with two stages of amplification of the corrective impulse given by the governor, the advantage being a higher sensitivity of the entire controlling system. Hydraulic systems with two-stage amplification have also constructional advantages, inasmuch as through-flow valves can be used for the first stage and cut-off type valves for the second.

The through-flow type of valve can only be used with servo-mechanisms which are subjected to light loads; the impulses can then be led to any point in the control system of the turbine. Control systems of this type are used on turbines at the Stalin Factory and on many manufactured in other countries than the U.S.S.R.

A method for the rapid calculation of such systems will now be developed. Although various assumptions have to be made in solving the differential equations, the calculation gives useful results and makes it possible, not only to assess the effect of the different factors on the control, but also to determine the character of the speed variation of the turbine.

The system shown in Fig. 1, assuming a governor of negligible inertia, can be represented by the following differential equations:—

$$\left.\begin{array}{ll} 1. & \text{Turbine} \\ & T_a\,\varphi' - \mu_1 = 0 \\ 2. & \text{Main pilot valve} \\ & T_1\,\mu_1' + \mu_1 - \mu_2 = 0 \\ 3. & \text{Auxiliary pilot valve} \\ & T_2\,\mu_2' + \mu_2 + (\varphi/\delta) = 0 \end{array}\right\} \quad .. \quad (1)$$

where the dash signifies d/dt; T_a, T_1, T_2 = time constants of the turbine, and main and auxiliary pilot valves; φ = relative variation in turbine angular velocity; δ = degree of irregularity of controlling process; μ_1, μ_2 = relative positional variation of pilot valves.

The determinant of the simultaneous system (1), equated to zero, is

$$\begin{vmatrix} T_a\,\omega & -1 & 0 \\ 0 & T_1\,\omega+1 & -1 \\ 1/\delta & 0 & T_2\omega+1 \end{vmatrix} = 0$$

and can be expanded into

$$\omega^3 + \frac{T_1 + T_2}{T_1\,T_2}\,\omega^2 + \frac{\omega}{T_1\,T_2} + \frac{1}{\delta\,T_a\,T_1\,T_2} = 0 \quad .. \quad (2)$$

Introducing

$$\tau = \delta\,T_a, \quad k = T_1/\tau, \quad n = T_2/\tau,$$

we have

$$\omega^3 + \frac{k+n}{kn\tau}\,\omega^2 + \frac{\omega}{kn\tau^2} + \frac{1}{kn\tau^3} = 0 \quad .. \quad (2a)$$

and with the new variable $z = \tau\omega$,

$$z^3 + \frac{k+n}{k\,n}\,z^2 + \frac{z}{kn} + \frac{1}{kn} = 0 \quad .. \quad (2b)$$

It will be noted that eq. (2b) is symmetrical in k and n, and consequently both pilot valves have equal effects on the controlling process. In practice, it is advisable to design the auxiliary pilot valve with a small time constant, since its oil requirement will be small, even with a low value for T_2, in view of the small amount of power transmitted. Normally, the controlling process has a damped periodic character, and the characteristic equation of (2b) will then have one real root z_1, and two complex roots $z_2, z_3 = \alpha \pm i\beta$.

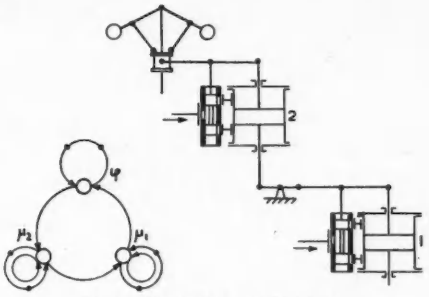


Fig. 1. **Standard control system.**

The real root is

$$z_1 = u + v - [(k+n)/(3kn)],$$

while the components of the complex roots are

$$\alpha = [(u+v)/2] - [(k+n)/(3kn)],$$

and

$$\beta = \sqrt{3}\,(u-v)/2,$$

where

$$u = \sqrt[3]{-q + \sqrt{q^2 + p^3}}$$

and

$$v = \sqrt[3]{-q - \sqrt{q^2 + p^3}},$$

p and q being defined by

$$p = (b/3) - (a^2/9),$$
$$q = (a^3/27) - (ab/6) + (c/2).$$

a, b, and c are the coefficients of z in eq. (2b). Substituting for a, b and c their values from eq. (2b) gives

$$p = -[k^2 - kn + n^2]/[9k^2n^2]$$
$$q = [2\,(k^3+n^3) - 3kn(k+n)]/[54k^3n^3] + 1/(2kn)$$

The equations for the parameters determining the controlling process can now be written as follows:—

1. Turbine:

$$\varphi = A\,e^{z_1\,t/\tau} + e^{\alpha\,t/\tau}\,[B\cos(\beta t/\tau) + C\sin(\beta t/\tau)] \quad (3)$$

2. Main pilot valve: The equation for positional variation can be obtained from the condition $\mu_1 = T_a\varphi'$ or $\delta\,\mu_1 = \tau\varphi'$, i.e. by differentiating eq. (3):

$$\delta\,\mu_1 = Az_1\,e^{z_1\,t/\tau} + e^{\alpha\,t/\tau}\,[(\alpha\,B + \beta\,C)\cos(\beta\,t/\tau) + (\alpha\,C - \beta\,B)\sin(\beta\,t/\tau)] \quad .. \quad (4)$$

3. The piston displacement of the main pilot valve is therefore determined by $\sigma_1 = T_1\,\mu_1'$

$$\sigma_1 = \frac{T_1}{\delta\tau}\left\{ Az_1^2\,e^{z_1\,t/\tau} + e^{\alpha\,t/\tau}\,[(\alpha^2\,B + 2\,\alpha\,\beta\,C - \beta_2\,C)\cos(\beta t/\tau) + (\alpha^2\,C - 2\,\alpha\,\beta\,B - \beta^2\,C)\sin(\beta t/\tau)] \right\} \quad .. \quad .. \quad .. \quad (5)$$

The constants of integration will now be determined. When $t = 0$, in eq. (3), $\varphi_0 = -\delta\lambda = A + B$, where φ_0 = velocity variation from the value corresponding to the new turbine load, δ = degree of irregularity, λ = measure of sudden load change. Furthermore, when $t = 0$, the variation of the main pilot valve is $\mu_{01} = \lambda$. This gives a second condition for the constants:

$$Az_1 + \alpha\,B + \beta\,C = \lambda.$$

Finally, when $t = 0$ the velocity of the pilot valve is zero and therefore the variation of the piston in the main pilot valve is also zero. Eq. (5) then gives

$$Az_1^2 + B(\alpha^2 - \beta^2) + 2\alpha\beta C = 0$$

hence $A = A_1\,\delta\lambda$; $B = B_1\,\delta\lambda$; $C = C_1\,\delta\lambda$.

The equations for the controlling process become

$$\left.\begin{aligned}
\varphi &= \delta\lambda\,[A_1 e^{z_1 t/\tau} + e^{\alpha t/\tau}(B_1\cos(\beta t/\tau) \\
&\quad + C_1 \sin(\beta t/\tau))] = \delta\lambda\,.\,f_1(n, k, t/\tau), \\
\mu_1 &= \lambda\,\{A_1 z_1 e^{z_1 t/\tau} \\
&\quad + e^{\alpha t/\tau}[(\alpha B_1 + \beta C_1)\cos(\beta t/\tau) \\
&\quad + (\alpha C_1 - \beta B_1)\sin(\beta t/\tau)]\} \\
&= \lambda\,.\,f_2(n, k, t/\tau), \\
\sigma_1 &= \frac{T_1}{\delta\tau}\{A_1 z_1^2 e^{z_1 t/\tau} + e^{\alpha t/\tau}[(\alpha^2 B_1 \\
&\quad + 2\alpha\beta C_1 - \beta^2 B_1)\cos(\beta t/\tau) \\
&\quad + (\alpha_2 C_1 - 2\alpha\beta B_1 - \beta^2 C_1) \\
&\quad \sin(\beta t/\tau)]\} = \frac{k}{\delta}\,.\,f_3(n, k, t/\tau).
\end{aligned}\right\} \quad .. \quad (6)$$

The equations for the determination of the constants are:

$$\left.\begin{aligned}
A_1 + B_1 &= -1, \\
A_1 z_1 + B_1\alpha + C_1\beta &= 1, \\
A_1 z_1^2 + B_1(\alpha^2 - \beta^2) + 2\alpha\beta C_1 &= 0,
\end{aligned}\right\} \quad .. \quad (7)$$

therefore

$$\begin{aligned}
B_1 &= [2\alpha(z_1 + 1) - z_1^2]/[(z_1 - \alpha)^2 + \beta^2], \\
A_1 &= -1 - B_1, \\
C_1 &= [1 + z_1 + B_1(z_1 - \alpha)]/\beta.
\end{aligned}$$

The constants A_1, B_1, and C_1, are functions only of the relative times k and n. If time is taken as the independent variable, the expressions in the brackets, which determine the changes in angular velocity and the piston motion of the pilot valves, can be regarded as universal functions dependent only on the relative times of the valves and on t/τ. These functions can be calculated for different values of k and n. This simplifies the subsequent determination of the controlling process required for any given practical case.

In designing a control system it is most important to determine the maximum amplitude of the velocity variation under the most unfavourable conditions, i.e., when the load is varied from full load to zero load. Then

$$\left.\begin{aligned}
(\Delta n/n)_{\max} &= \delta\lambda + \varphi_{\max} \\
&= \delta\lambda + \delta\lambda\, f_{1\,\max}(n, k, t/\tau) \\
&= \delta\lambda\,[1 + f_{1\,\max}(n, k, t/\tau)]
\end{aligned}\right\} \quad (8)$$

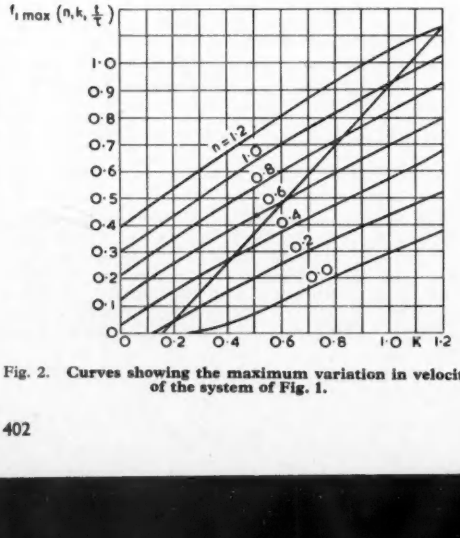


Fig. 2. **Curves showing the maximum variation in velocity of the system of Fig. 1.**

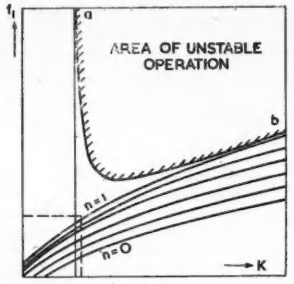


Fig. 3. **Range of instability of the system of Fig. 1.**

$f_{1\,\max}$ can be found by equating f_2 to zero and finding t/τ. Introducing then t/τ into the expression for $f_1(n, k, t/\tau)$ we obtain $f_{1\,\max}$ and from eq. (8) the maximum speed variation $\varphi_{\max}$ when the turbine load is dropped to zero. The curves in Fig. 2 have been computed by this method. These curves considerably simplify the calculation of control systems with two stages of amplification. This will be illustrated by an example.

It is required to find the increase in turbine speed when the load is dropped from full load to zero load, assuming that $T_a = 10$ sec, $\delta = 4$ per cent, $T_1 = 0{\cdot}32$ sec and $T_2 = 0{\cdot}08$ sec. This gives $\tau = \delta T_a = 0{\cdot}4$, $k = T_1/\tau = 0{\cdot}8$, and $n = T_2/\tau = 0{\cdot}2$. If the no-load power is 7 per cent of the rated output, then dropping the load corresponds to $\lambda = 0{\cdot}93$. From the curves of Fig. 2, $f_{1\,\max} = 0{\cdot}34$, and the increase in speed is given by $(\Delta n/n)_{\max} = 0{\cdot}93 \times 0{\cdot}04\,(1 + 0{\cdot}34) = 0{\cdot}05$. The diagram Fig. 2 is drawn for values of n and k between 0 and 1·2, which are most frequently encountered in practice. Fig. 3 shows the same curves for a wider range of values of k. The dotted square in the left-hand corner of Fig. 3 represents the region covered by Fig. 2, and it is evident that the controlling systems used in practice are at a safe distance from the area of instability (shaded area) shown in Fig. 3.

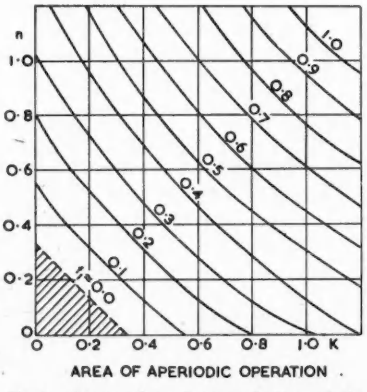


Fig. 4. **Curves of maximum velocity variation.**

Fig. 4 shows the same relationship as Fig. 2, but with f_1 as parameter. The shaded area corresponds to those values of n and k for which the dynamic rise in speed does not exceed the static deviation from rated speed.

Systems with two pilot valves are usually more sensitive than systems with a single pilot valve, because the centrifugal force of the governor can be smaller, and, therefore, a governor with a smaller moment of inertia can be used. Sluggishness in operation is undesirable, particularly since it increases with in-

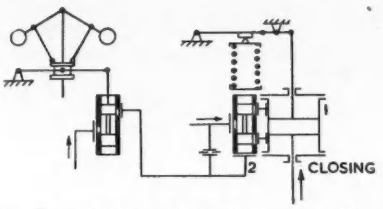


Fig. 5. **Control system using spring compensation.**

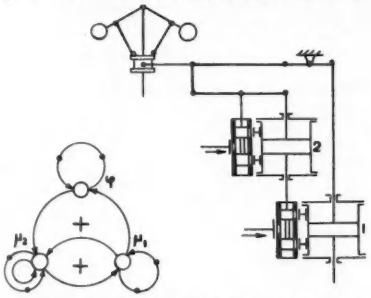


Fig. 6. **Modification of the system of Fig. 5.**

creased wear of the governor components, and may not easily be noticed, especially when several machines operate in parallel.

In the case of a turbine for which $T_a = 10$ sec, $\delta = 0{\cdot}5$, and $T_1 = 0{\cdot}5$ sec, the sensitivity error will be $\epsilon = 0{\cdot}6$ per cent. If a second pilot valve is introduced, ϵ can be reduced to 0·2 per cent.

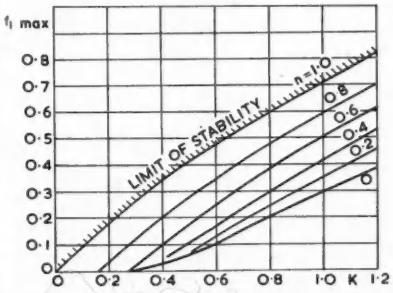


Fig. 7. **Curves giving maximum velocity variation for the system of Figs. 5 and 6.**

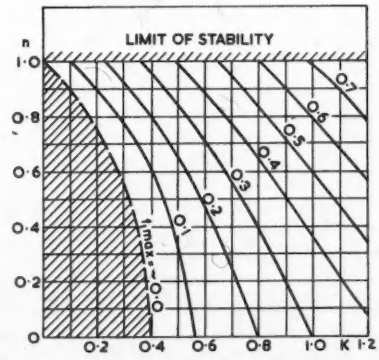


Fig. 8. **Curves of maximum permanent velocity variation for the systems of Figs. 5 and 6.**

If in a first approximation it is assumed that the speed rise when the load is dropped is proportional to the error in sensitivity, then from Fig. 2, assuming $\lambda = 0{\cdot}97$ for a system with one pilot valve: $(\Delta n/n)_{max} = \delta\lambda(1 + f_{1\,max}) + \epsilon = 0{\cdot}93 \times 5(1 + 0{\cdot}29) + 0{\cdot}6 = 6{\cdot}6$ per cent. For a system with two pilot valves and a time constant $T_2 = 0{\cdot}05$ sec for the second valve: $(\Delta n/n)_{max} = 0{\cdot}93 \times 5(1 + 0{\cdot}36) + 0{\cdot}2 = 6{\cdot}5$ per cent.

The control system shown in Fig. 5 is gaining ground in present-day practice. In the first-stage pilot valve, a through-flow of oil is obtained, whereas the main pilot valve (2) is of the cut-off type. Fig. 6 is a modification of the system shown in Fig. 5.

The system of differential equations for a controlling system of the type shown in Figs. 5 and 6 is as follows:

1. Turbine
$$T_a\,\varphi' - \mu_1 = 0,$$
2. Main pilot valve
$$T_1\,\mu_1' - \mu_2 = 0,$$
3. Auxiliary pilot valve
$$T_2\,\mu_2' + \mu_2 + \mu_1 + (\varphi/\delta) = 0.$$

The determinant of the system, equated to zero, is

$$\begin{vmatrix} T_a\omega & -1 & 0 \\ 0 & T_1\,\omega & -1 \\ 1/\delta & +1 & T_2\,\omega + 1 \end{vmatrix} = 0$$

and this gives the characteristic equation

$$\omega^3 + \frac{\omega^2}{T_2} + \frac{\omega}{T_1\,T_2} + \frac{1}{\delta\,T_a\,T_1\,T_2} = 0. \quad .. \quad (9)$$

As previously $\tau = \delta T_a$, $k = T_1/\tau$, $n = T_2/\tau$, $z = \tau\omega$, so that we have

$$z^3 + (z^2/n) + (z/nk) + (1/nk) = 0. \quad .. \quad (9a)$$

In this equation the time constants n and k are not symmetrical coefficients. The parameters determining the roots of the characteristic equations are

$$p = (3\,n - k)/(9\,kn^2);\; q = [(2\,k - 9\,n)/(54\,n^3\,k)] + 1/(2\,kn).$$

All further transformations leading to the final equation for the turbine and the control elements will be similar to those already obtained. The maximum rise in velocity, φ_{max}, is given in Figs. 7 and 8.

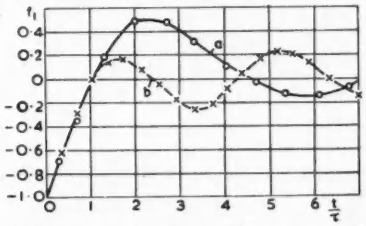


Fig. 9. **Velocity variation of two different types of control systems.**

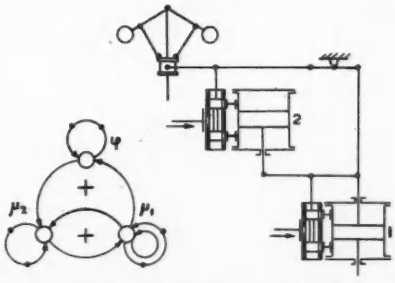


Fig. 10. **Modification of the control systems of Figs. 5 and 6.**

A comparison of the control systems of Fig. 1 and Figs. 5 or 6 shows that in the latter systems the rise in speed is smaller if k and n are equal, and also that the limit of instability for these systems is closer to the normal operating range. Thus, in the system of Fig. 5 it becomes necessary to keep the time-constant of the main pilot valve as low as possible. Fig. 7 shows that whereas a larger time constant k for the main pilot valve will not cause instability, a time-constant $n = 1{\cdot}0$ for the auxiliary pilot valve will produce undamped oscillations.

Fig. 9 gives the speed variation for the system of Fig. 1 in curve a, and for the systems of Figs. 5 and 6 in curve b. Both curves are plotted for $n = 0{\cdot}8$ and $k = 0{\cdot}6$.

The characteristic equation (9a) contains in its second term only n, the constant for the auxiliary pilot valve. If, in this equation, the coefficients k and n are interchanged, the control system to which such an equation applies is represented by Fig. 10, and corresponds to the characteristic equation

$$z^3 + (z^2/k) + (z/nk) + (1/nk) = 0.$$

This system is not practical since it requires small values of k to obtain adequate stability, whereas n will have little influence on stability. The reduction of k can only be obtained, however, by increasing the output of the governor, and, therefore, the systems of Fig. 5 or 6 are superior.

The Contact Pressure in Spur Gearing

By H. PROTAT. (From *Machines et Métaux, France*, Vol. 32, No. 359, July, 1948, pp. 235-238.)

A GRAPHIC method of determining the contact pressure in a pair of spur gears is presented.

Consider a pair of external gear wheels (Fig. 1), the pinion of which has its centre at O_1 and the gear its centre at O_2, with the pitch point at A. The following notation may be used:—

φ = pressure angle,
p = circumferential pitch,
l = width of contact,
ρ_1 = radius of curvature of flank of pinion, at N,
ρ_2 = radius of curvature of face of gear, at N,
E_1 = Young's modulus of pinion material,
E_2 = Young's modulus of gear material,
F = tangential force at pitch point,
F_n = normal force on flanks of teeth $= F/\cos\varphi$,

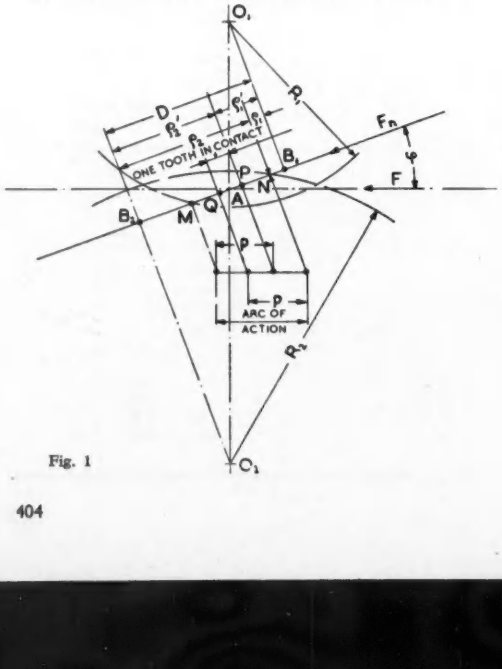


Fig. 1

f_n = normal force per unit length $= F_n/l$,
n = contact pressure (consistent units throughout).

Consider the point N, on the line of action. Hertz's formula gives:

$$n = \sqrt{\frac{E_1 E_2}{0{\cdot}9\,\pi\,(E_1 + E_2)}}\sqrt{P}$$

where $P = f_n\left(\frac{1}{\rho_1} \pm \frac{1}{\rho_2}\right)$ and the positive sign is used for external gears, the negative being used for internal gears.

Now let $D = B_1 B_2$, the distance between the projections of O_1 and O_2 on the line of action;

for external gears: $\rho_2 + \rho_1 = D$ (see Fig. 1.)

for internal gears: $\rho_2 - \rho_1 = D$ (see Fig. 2).

Hence $P = f_n D/\rho_1 \rho_2$ for both external and internal gears. The ambiguity of sign is thus eliminated from the expression for P, and the three variables D, ρ_1 and ρ_2 are directly measurable from the layout of the gears.

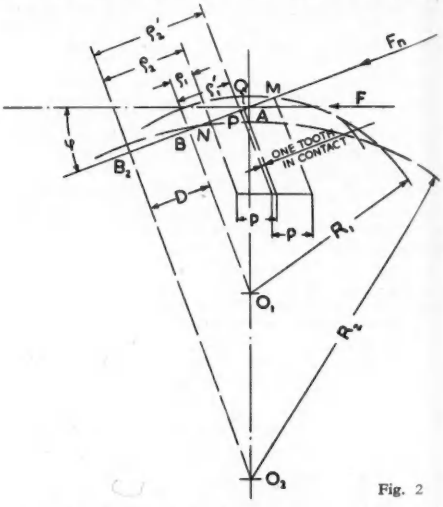


Fig. 2

It should be noted that P increases as the product $\rho_1 \rho_2$ decreases. Note also that the following observations on external gears equally apply to internal gears.

Consider the case where not more than two pairs of teeth are in contact simultaneously. Since

$$MP = NQ = p\cos\varphi,\ MN/p\cos\varphi \leqslant 2.$$

Over the length PQ of the line of action, only one pair of teeth are in contact. Over the remaining parts, two pairs are in contact. In both cases the product $\rho_1 \rho_2$ is smallest at the point nearest to B_1. Hence the critical contact pressure occurs at P or at N.

Let ρ_1' and ρ_2' be the radii of curvature at the point P; at that point the quantity P is then given by:

$$P_1 = F_n D/l\rho_1' \rho_2'.$$

At the point N the normal force is carried by two teeth, and therefore:

$$P_2 = F_n D/2l\rho_1 \rho_2.$$

The higher of these two values of P is substituted in Hertz's formula to give the maximum contact pressure.

In the special case of a rack and pinion, $D = \rho_2 = \infty$ and the expression becomes:

$$P = f_n/\rho_1.$$

The Value of the Heat Pump for Heating Installations

By E. WIRTH. (From *Sulzer Technical Review*, No. 1, 1948, pp. 29-41, 16 illustrations.)

It was neither lack of scientific knowledge nor of machines capable of applying such knowledge which prevented the heat pump from being widely adopted at an earlier date. The real obstacle was the familiar fact that the impetus towards the practical employment of a superior method can only be provided by suitable economic conditions. The generation of artificial cold had long been an urgent requirement of the food and brewing industries, and it was for this reason that the necessary machines were invented and built so soon. Heat, however, was obtainable from other sources than the heat pump. The most obvious means of heat generation is combustion, and fuel-fired boilers with good efficiencies were available long before the heat pump was invented. Nor was there any lack of fuels, and these were indeed so cheap that some of them, particularly coal, were used in a very wasteful manner. There was, therefore, no obvious reason for replacing the simple coal-fired boiler by a machine such as the heat pump, which was for those days a fairly complicated proposition.

Today, however, coal is no longer a fuel alone, but is becoming a more and more important raw material for the chemical industry. Under these circumstances we have every reason for making the very best use of our coal and for seeking other and more economical methods of generating heat.

Before the various applications of the heat pump are explained, a simple pictorial presentation will be useful as an illustration of the analogy between hydraulic and thermal processes.

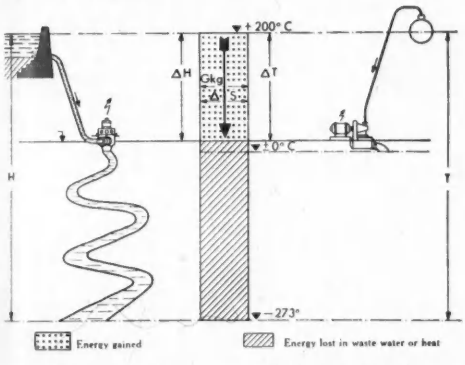


Fig. 1. **Generation of electric energy with water power (on left) or heat (on right) by allowing it to flow downwards in the direction of the black arrow.**

Fig. 1 shows simplified diagrams of a waterpower plant on the left and a steam-power plant on the right. The water level of the hydraulic storage lake at the top left may be assumed to lie at a height of 473 metres above sea level, and the tail race of the water turbine at 293 metres. The water leaving the turbine enters a river which finally flows down to the sea. The hydraulic energy supplied to the water turbine corresponds to the product of the effective head ΔH [m] and the weight of water G [kg]. The energy balance of this hydraulic plant is shown in the middle of Fig. 1, where the weight of the water is plotted along the abscissae and the head along the ordinates. The dotted area represents the work theoretically available for use, the hatched area the energy not utilized but lost through flow resistances in the river.

The steam boiler visible at the top right may be assumed to supply steam at 200° C (or 473° C absolute temperature), which is expanded in the condensing steam turbine below it to, say, 20° C (or 293° C absolute temperature). These figures are chosen to correspond to those in the hydraulic example. The turbine drives an electric generator. The theoretically useful energy corresponds to the product of the available temperature drop ΔT °C and the entropy difference ΔS which may for explanatory purposes, if not quite accurately, also be termed the "heat weight." The entropy itself is defined by the formula

$$S = \int \frac{dQ}{T} \text{ kcal°C}^{-1}.$$

The energy balance can be represented graphically and, with the temperature and entropy figures chosen, gives exactly the same diagram as the hydraulic power plant. The dotted area again corresponds to the theoretically useful work, while the hatched area is the energy loss inevitably involved, because the heat supplied cannot be cooled down to the absolute zero of −273° C, but leaves the steam turbine at the fairly high absolute temperature of $T = 293°$ C.

This presentation of the energy balance sheet is nothing else but the schematic entropy diagram of the thermal energy conversion. As the entropies are plotted along the abscissae and the absolute temperatures along the ordinates, the areas which appear as the product of these magnitudes represent heat quantities. The water and heat generate power while flowing in the direction of the black arrow from the upper to the lower altitude or temperature level. On account of its simplicity, this method of representation is also used in the following examples.

While Fig. 1 illustrates the analogy between hydraulic and thermal energy production, Fig. 2 shows the energy conversion in the pumping of water and heat. The water and the heat are raised from the lower to the higher level in the direction of the white arrow, energy being expended in the process. The pumping plant on the left raises the water from a height of 293 to 473 metres above sea level, to retain the figures used in the first example. The theoretical energy requirements are represented by the dotted area on the left. If it were desired to represent the energy content of the water

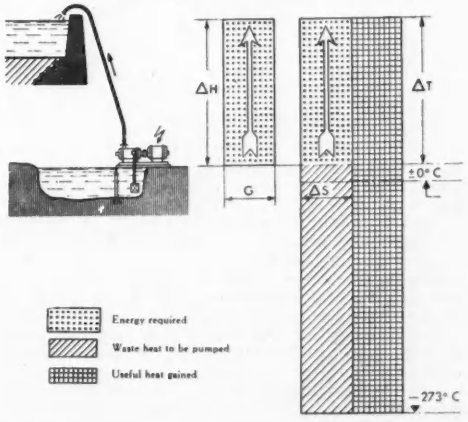


Fig. 2. **Pumping water (on left) or heat (on right) upwards in the direction of the white arrow.**

before pumping as compared with sea level, the strip would have to be extended below the dotted area down to sea level. This would only have a theoretical significance, however, as this energy is not practically in evidence in the present case.

In the pumping of heat a definite heat quantity, represented by the diagonally hatched area, is absorbed from the surroundings (air or watercourse) at 20° C (or 293° C abs.) by a somewhat cooler vaporiform or gaseous working medium, after which the working medium is compressed. The work theoretically required for this purpose in the compressor is shown in the dotted area on the right. The compression brings the working medium, with the heat absorbed by it, up to a temperature of 200° C (or 473° C absolute). The heat equivalent of the mechanical energy introduced and the quantity of heat pumped to a higher temperature are represented together by the squared rectangle on the right. This heat quantity, as can be seen, is considerably greater than the heat equivalent of the mechanical energy introduced. What is more, as a result of its higher temperature it can be used, for instance, in a central heating system. The ratio of the useful heat (squared area) to the energy introduced (dotted area) is known as the coefficient of performance, or performance figure. This is a measure of the efficiency of a given heat pump process and plays an important part in any assessment of its economy. It is now at once clear that the performance figure will be the higher, the lower the temperature difference ΔT by which the heat from the surroundings must be raised, and in a lesser degree the higher the initial level, i.e., the temperature at which the heat is drawn from the surroundings by the heat pump.

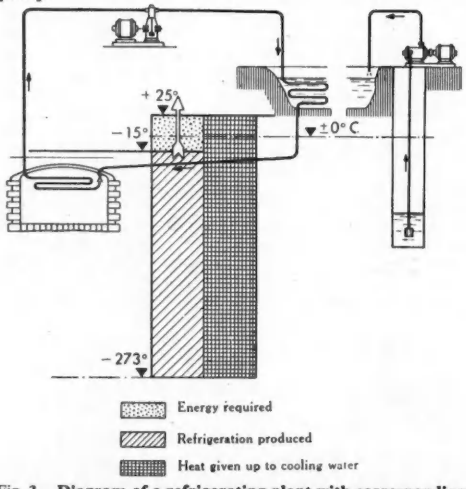


Fig. 3. Diagram of a refrigerating plant with corresponding heat diagram. To the right a pump for purposes of hydraulic analogy.

Fig. 3 shows the basic arrangement of an orthodox refrigerating plant with reciprocating ammonia compressor and its heat balance. The heat is extracted from the cellar at — 15° C, is pumped up to + 25° C by the compressor and at this temperature is given up to the water in the top right-hand corner. To the right a pump plant used for keeping down the water level in building foundations serves as a hydraulic analogy.

The heat given up at + 25° C is mostly too low in temperature to be capable of practical employment. Theoretically, however, it is quite possible to raise the final pressure of the compressor and thereby to bring the heat up to a temperature level at which it can be utilized. The diagram of such a plant is given in Fig. 4, and to the right of it the corresponding hydraulic process. The heat is here extracted from a watercourse at + 2° C, is pumped up to + 70° C and is given up with the thermal equivalent of the energy introduced to the central heating systems shown schematically in the diagram. Plants of this kind are known as heat pumps.

The entropy diagrams used here to illustrate the energy balance correspond to the ideal Carnot cycle. As far as the heat pump is concerned, the figures practically attainable are only about 50 or 60 per cent of the theoretical ones. Modern heat pump plants nevertheless give effective performance figures between 3 and 6. This means that the 860 kcal, which is the heat equivalent of 1 kWh, together with the heat pumped from the surroundings, permits a useful heat quantity of 2,500-5,000 kcal to be given off. In evaporating plants the values attained may be considerably higher.

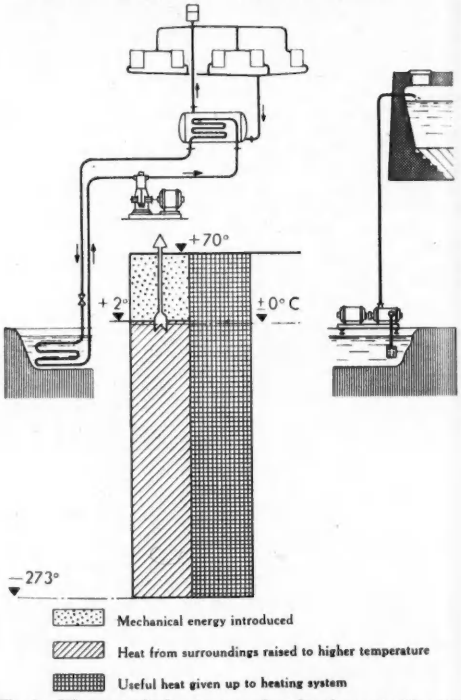


Fig. 4. Diagram of a heat pump plant for the central heating of a building, with corresponding heat diagram. To the right a hydraulically analogous pumping plant.

The economy of heat pump plant naturally depends also on interest, depreciation, etc., and it is, therefore, desirable that the working hours per year should be as high as possible. This requirement is met, to a large extent, by combined plants used both for cooling and heating, the heat diagram of which is shown in Fig. 5. The lower compression stage brings the heat obtained at — 17° C up to + 22° C (71° F). This heat quantity, to which is added the thermal equivalent of the energy introduced, is then raised in the upper stage of compression from + 22° C to + 65° C and, again augmented by the heat value of the energy supplied, is used for heating. An analogous two-stage pumping plant is shown on the left, one of the pumps serving chiefly for suction while the other lifts the liquid to a tank at a higher level.

A combined plant working within these tempera-

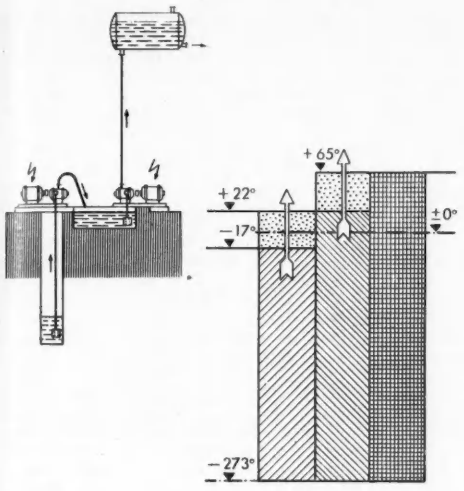


Fig. 5. **Diagram of a heat pump used simultaneously for heating and cooling, with corresponding heat diagram. To the left a hydraulically analogous two-stage pumping plant.**

tures was actually installed in a large industrial undertaking in Switzerland in the period of fuel shortage during the war. A refrigerating plant consisting of three compressor units was already in existence, and to this a further compressor was added to act as an upper stage. The plant can be finely regulated over a wide range and with very little loss. As refrigeration is required in any case and the utilization of the heat is effected exclusively through the additional upper stage, the performance figure can refer to the high-pressure stage alone. The measurements made for this purpose have shown that each kilowatt-hour used for driving the upper compressor stage produces 4,700 kcal of useful heat, which corresponds to a performance figure of 5·45.

Another interesting application for a combined plant is that of a swimming bath which is converted in winter into a skating rink. For both of these the same heat pump can be used, the lower temperature stage serving for the ice rink and the upper stage for the heated swimming bath.

These combined plants can also be used alternately for heating and cooling. A specially suitable system is then panel heating, as it requires lower initial temperatures than radiators, while the same wall or ceiling surface can be used for radiant heating in winter and for cooling in summer.

A good example of this type of plant is a heat pump erected in an industrial building with a volume of 350,000 cu ft. The same radiation piping system is used for cooling and heating throughout the year. Heat is extracted from the works water supply, which is cooled by about 1° C by the heat pump. A new regulating system involving only very slight losses enables the output of the pump to be adjusted at all times so that the initial temperature in the heating system is kept constant. The transition from heating to cooling service is effected by a simple switching operation, the heating water then being conducted to the evaporator and the mains water to the condenser.

Under favourable conditions, however, the heat pump may also be advantageous when its only duty is the supply of heat, as the following example will show.

A heat pump plant of a capacity of 640,000 B. Th. U. per hour is at present being built for the administrative building of a large industrial enterprise. This heat pump is to be used only for space heating purposes, but in view of its favourable layout and high loading at all times it can be regarded as extremely economical. The source of heat is in this case comparatively warm subsoil water which is pumped from a shaft next to the machine room and conducted to the evaporator. As there is an existing panel heating installation, the water has to be heated to 40-45° C only. In spring and autumn the heating output of the pump exceeds the demand, and the surplus is then used for preheating the return water of a radiator heating system, the heat pump being regulated in accordance with the requirements of the latter. If the temperature of the radiator return water exceeds 40° C, the radiator heating system is again disconnected from the panel warming system and is operated with its own fuel-fired boiler. As soon as the outside temperature falls below + 2° C, the water of the panel warming system leaving the heat pump is passed on to a boiler for further heating. Regulation is automatic throughout. Under these circumstances the heat pump is at all times under load and is, therefore, able to cover 90 per cent of the total heat requirements. Although it is restricted to winter service, the full working hours per year amount to over 3,000. The mean performance figure is about 4.

Another noteworthy heat pump also used for heating purposes only was recently installed in a Swiss weaving and cotton spinning mill. The heat produced is used in air conditioning plants and for air heaters in the work-rooms. The maximum requirements are 2,200,000 B. Th. U. per hour, half of which can be supplied by the heat pump plant. The remainder is provided chiefly by an oil-fired heating boiler, while peak loads are taken by the existing steam plant. Nevertheless, over 90 per cent of the heat required annually is supplied by the heat pump. Its source is the water leaving the factory's turbine, this being cooled by about 1 or 2° C.

A combined plant which works very economically owing to the high number of operating hours and favourable temperatures was supplied to a large cotton mill in Shanghai. The heat pump serves an air conditioning plant and is used either for heating or cooling according to the season. The heat is drawn from the water of a nearby river, which never falls below 10° C even in winter. The maximum refrigerating requirements in summer are in excess of the maximum heat requirements in winter. Skilful combination has enabled these requirements to be fulfilled on highly economical lines with the same heat pump, which operates for practically 24 hours per day throughout the year. The total number of operating hours per year is therefore above 7,000, while the performance figure is over 4.

Fig. 6 illustrates the employment of a heat pump in a drying plant. The air leaving the drying chamber carries a good deal of moisture, but is still warm. It is cooled in the evaporator of the heat pump and loses most of its water by precipitation. After being dried in this way the air is again heated in the condenser at 40° C and then returned to the drying chamber by the circulating fan.

Heat pump plants of this kind have also been built for underground magazines, which have to be kept dry. The air is freed of moisture by undercooling and then heated by passage through the heat pump. This air circuit permits the moisture penetrating into the magazines through the surrounding rock to be satisfactorily eliminated.

A drying plant for the earthenware industry is at present under construction. It will be in service throughout the year, and the performance figure will in this case be approximately 5.

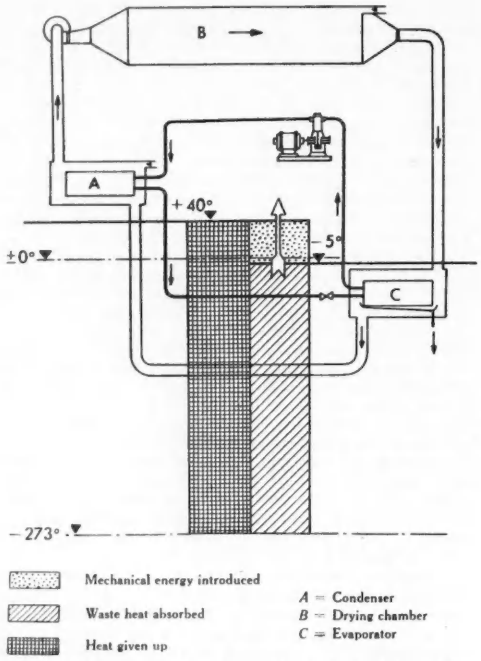


Fig. 6. **Diagram of a heat pump for a drying plant, with corresponding heat diagram.**

It may now be interesting to examine the diagram obtained with a heat pump driven by a heat engine. Actual figures for this case are available in the plant supplied by Sulzer Brothers in 1879. The heat engine consists of a drop-valve steam engine receiving its steam at 160° C and giving it up after expansion as process steam at 110° C, see Fig. 7. The refrigerating machine driven by the steam engine raises the heat obtained at — 15° C to + 25° C and gives it up to the cooling water. The first rectangle on the left in the heat diagram shown in Fig. 7 represents the heat introduced. The dotted upper area corresponds to the mechanical energy generated, with which the refrigerating machine is driven. The hatched area below it is the heat available for process uses. The second oblong from the left shows the mechanical energy absorbed by the refrigerating machine as a horizontally shaded area at the top and the refrigerating capacity as a hatched area below it. This is how the existing plant actually works.

If a urther compression stage had later been added to the refrigerating machinery, the heat given up by the first stage at + 25° C could have been raised to + 65° C and put to use in a heating system. An electric motor might have been used for the additional power requirements. The quantity of heat thus recovered is represented by the squared area on the right enclosed by a broken line. The total amount of heat introduced (100 per cent), which is represented by the first oblong on the left, and the 9 per cent of additional electric energy used would, in this case, have supplied about 89 per cent of useful heat for process purposes and 77 per cent for heating, making a total of 166 per cent, to which the useful refrigeration would still have to be added.

A modern plant of this kind is that installed in the district heating station of the Swiss Federal Institute of Technology in Zurich. The reciprocating ammonia compressor shown in Fig. 8 is driven by an electric motor. The heat is taken from the River Limmat. Here too the waste heat of the thermal power station is turned to account. When sufficient or surplus hydro-electric energy is available, this is employed primarily for driving the heat pump. In this way considerable quantities of fuel can be saved. When little hydro-electric energy is available, the heat pump is driven by the current produced in the Institute's thermal power station.

Obviously, a Diesel engine may also be used instead of a steam power plant for driving the heat pump, as here too the waste heat can be utilized with very little additional trouble. It is then quite possible to obtain an upper waste heat stage which is high enough for industrial purposes and a lower stage which can be used for space heating or for the preparation of hot water. The temperature of the waste heat does not have to be raised by artificial means, so that the normal maximum output of the Diesel engine is fully available.

Fig. 9 shows the advantages secured with the combination of a Diesel engine and a heat pump. The calorific value of 1 kg of oil is compared with the useful heat which can be obtained by direct firing in a boiler and alternatively by employment in a Diesel engine with waste heat utilization and with heat pumps of various performance figures. As can be seen, the Diesel engine and the heat pump driven by it together

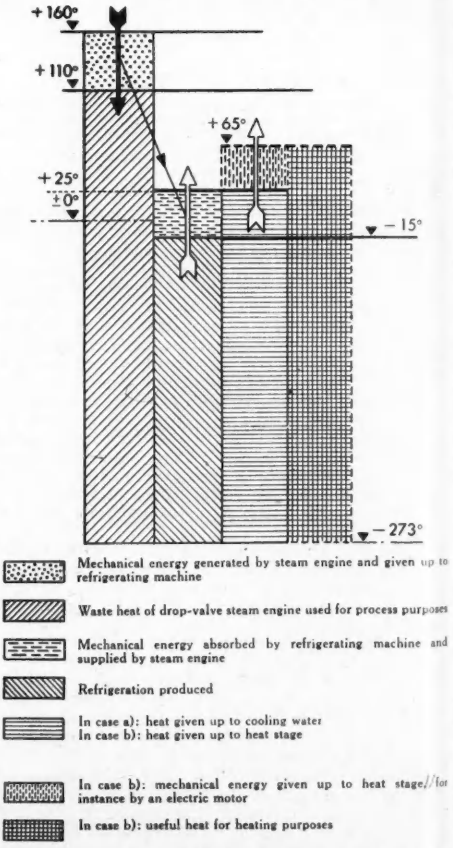
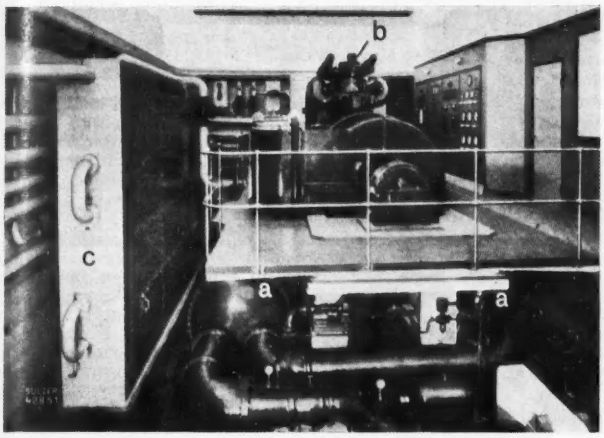

+ 160°
+ 110°
+ 65°
+ 25°
± 0°
– 15°
– 273°
Mechanical energy generated by steam engine and given up to refrigerating machine
Waste heat of drop-valve steam engine used for process purposes
Mechanical energy absorbed by refrigerating machine and supplied by steam engine
Refrigeration produced
In case a): heat given up to cooling water
In case b): heat given up to heat stage
In case b): mechanical energy given up to heat stage, for instance by an electric motor
In case b): useful heat for heating purposes

Fig. 7. **Schematic heat diagram of heat pump driven by a heat engine (a) based on the existing refrigerating plant (with full bounding line) and (b) on the assumption of an additional heat stage (with broken bounding line).**



(a) Evaporators through which water of River Limmat flows and in which the heat contained therein is absorbed by the heat pump.
(b) Automatically regulated reciprocating ammonia compressor.
(c) Condenser of heat pump serving as heating boiler for outgoing water of central heating system.

Fig. 8. Reciprocating ammonia compressor with automatic regulation forming part of the heat pump in the district heating works of the Swiss Federal Institute of Technology in Zurich.

supply more useful heat for heating purposes than is contained in the fuel needed for the Diesel engine.

For plants of this kind, units of small to medium output are preferable, as large works, owing to their large quantities of waste heat, must be connected to a district heating system, which is not always justified. This fact rather favours a decentralized layout for such plants, especially as reasonable decentralization has in times of need invariably proved superior to exaggerated centralization.

Nowadays, Diesel-generator sets are being used on a growing scale for covering peak loads and for emergency current requirements. The working hours of such plants are, by their very nature, fairly low. It would be quite feasible to make use of the Diesel engine in such plants for operating a heat pump, provided that over and above the heat or refrigeration produced, the exhaust heat of the engine could also be made use of. The working hours of the Diesel engine would in this way be increased. Experience with combined plants has shown that the time during which the Diesel engine would be needed for covering peak loads or generating emergency current—and the heat pump would consequently have to be shut down—could be bridged in the majority of cases with the aid of suitable heat and refrigeration accumulators. This may be an advantageous scheme, for instance, in department stores, which require large quantities of heat in winter and have to be cooled as effectively as possible in summer.

There are inevitably times when electricity supplies fail or have to be restricted, and this occurs chiefly in winter, precisely when the heat provided by the heat pump is at its most valuable. The solution is often the provision of a Diesel engine as well as an electric motor for driving the heat pump. A useful arrangement is shown in Fig. 10. The electric motor-generator 1 is coupled through disconnectable clutches 2 and 3 to a heat pump 4 and a Diesel engine 5. The heat pump can therefore be driven either by the electric motor or, with clutch 3 engaged, by the Diesel engine. When clutch 2 is disengaged, the motor-generator is driven by the Diesel engine and can then be used for generating current.

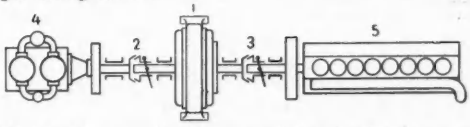


Fig. 10. Diagram of a heat pump driven by electric motor and Diesel engine.

It follows from the above considerations that the gas turbine can also be used for driving a heat pump if equipped with a waste heat utilization plant operating either direct or by way of the electric mains.

The examples cited and the considerations here put forward all go to support the statement that the heat pump is by no means a transitory product of the boom or a symptom of an " economy of shortage," but instead, wherever it is based on careful preliminary calculation, a valuable instrument of thermal economy for the future as well as for the present.

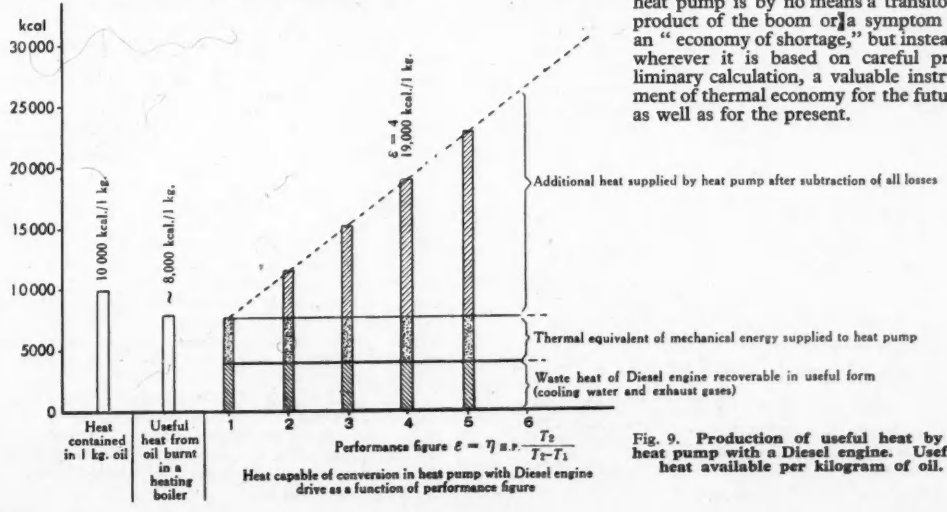

kcal
30000
25000
20000
15000
10000
5000
0
10 000 kcal./1 kg.
~ 8,000 kcal./1 kg.
ε = 4
19,000 kcal./1 kg.
Additional heat supplied by heat pump after subtraction of all losses
Thermal equivalent of mechanical energy supplied to heat pump
Waste heat of Diesel engine recoverable in useful form (cooling water and exhaust gases)
Heat contained in 1 kg. oil
Useful heat from oil burnt in a heating boiler
1
2
3
4
5
6
Performance figure ε = η u.p. T2/(T2−T1)
Heat capable of conversion in heat pump with Diesel engine drive as a function of performance figure

Fig. 9. Production of useful heat by a heat pump with a Diesel engine. Useful heat available per kilogram of oil.

CZECHOSLOVAKIA

On the Theory of Dielectrics

By J. Brablc. (From *Elektrotechnicky Obzor*, Vol. 36, No. 9, May, 1948, pp. 157-165, 17 illustrations.)

The properties of insulating materials differ from those of an ideal dielectric, not only on account of the loss angle and the dielectric polarization, which varies with frequency and temperature, but also on account of the dielectric absorption and the breakdown of the dielectric. Different theories have been put forward to explain these properties but none of these theories has general validity and experiments usually afford only qualitative proof. This paper describes measurements on oil-impregnated paper dielectric and attempts to give a theoretical explanation of its behaviour.

Existing Theories. The current flowing through a condenser is the resultant of a capacitive and a small ohmic component. Maxwell expressed this mathematically by introducing a complex expression for ϵ:—

$$\epsilon = \epsilon' - i\epsilon'' \quad .. \quad .. \quad .. \quad (1)$$

and for the loss angle:

$$\tan \delta = \frac{\epsilon''}{\epsilon'} \quad .. \quad .. \quad (2)$$

He describes a dielectric consisting of layers of materials of differing dielectric constants ϵ and conductivities λ. Wagner's dielectric with ϵ_1, λ_1, contains inhomogeneities ϵ_2, λ_2 of different shapes and distribution. Debye explains dielectric phenomena by considering the molecules as forming electric dipoles. All these theories result in expressions, where ϵ is complex, containing the frequency and, in Debye's theory, also the temperature. The general equation for ϵ is:

$$\epsilon = \epsilon' - i\epsilon'' = \epsilon_\infty \left(1 + \frac{k}{1 + \omega^2 \tau^2} - i \frac{k\,\omega\,\tau}{1 + \omega^2 \tau^2} \right) \quad (3)$$

$$\tan \delta = \frac{\epsilon''}{\epsilon'} = \frac{k\,\omega\,\tau}{1 + k + \omega^2 \tau^2} \quad .. \quad (4)$$

where ϵ_∞ . . . dielectric constant for $f \to \infty$

k, τ . . . values varying in accordance with different theories,

$\omega = 2\pi f$, and f = frequency.

In many cases, theoretically calculated values of ϵ and $\tan \delta$ could not be proved experimentally. However, the theories provided a qualitative explanation of observed facts, for instance the dielectric absorption.

Oil-impregnated paper dielectric is a combined system of several components, and the difficulties in explaining its properties at varying frequency and temperature are considerable. Most measurements ceased at the temperature of 80-90° C. The small peak of the loss angle at these temperatures is only an indication of much greater changes at temperatures in the region of 200° C. Measurements in the temperature range of 100° to 200° C lead to fresh views on the dielectric constant, loss angle and electric breakdown. The breakdown is a consequence of considerable changes in the dielectric constant and loss angle at these elevated temperatures.

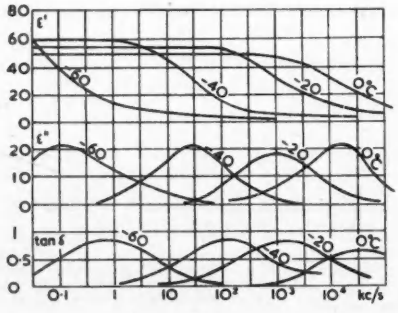


Fig. 1.

The Experiments. S. O. Morgan measured the properties of glycerin at varying temperatures and frequencies and his results can be seen in Fig. 1. ϵ' decreases with higher frequencies. The maxima of $\tan \delta$ correspond approximately to the points of inflection of the ϵ'' curves. Experiments on oil-impregnated paper resulted in curves of a different nature. These measurements were made within the temperature range of — 20° to 200° C and in the frequency range of 30 to 20000 c/s, using the Schering bridge. The voltage was 70 V, sinusoidal and constant at varying frequencies. A frequency generator was used with amplification before and after the bridge. The results are shown in the Figs. 2 — 5. The dielectric constant ϵ' increased as the temperature increased (Fig. 2), this being more pronounced at the lower frequencies. The increase was regular and slow up to 80° C, but reached the value of $\epsilon' = 12$ at higher temperatures, which is unusual for this dielectric. Fig. 3 shows ϵ' as a function of frequency. ϵ' decreases with increasing frequency and the curves present the shape of an inverted *S*. The points of inflection of these curves do not move to higher frequencies when the temperature increases, and the oil-impregnated paper differs in this property from glycerin.

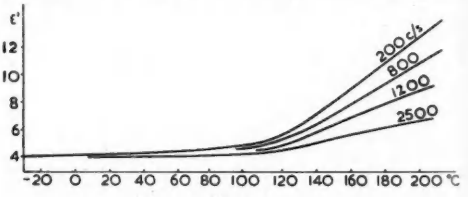


Fig. 2.

The inflection point is in the neighbourhood of 1000 c/s for oil-impregnated paper. The loss angle (Fig. 4) is also increasing to maximum values at about 200° C and the oil-impregnated paper could not be used in practice at so high a temperature. The oil-impregnated paper used for these measurements was in a single-core lead-sheathed cable. The coil-shaped cable was placed in an electric oven. Much of the oil was lost from the cable during the heating cycle; the loss-angle returned to 0·01, however, after cooling.

A comparison of these results points to the fact that glycerin (as well as water) and oil-impregnated paper are insulators of a differing nature. The change in the properties of glycerin will be explained by the orientation of the dipolar molecules. The change in the properties of the oil-impregnated paper might be understood with the aid of the inhomogeneity theory. However, eqs. (3) and (4) give no values for ϵ and $\tan \delta$ which can be verified. An attempt is made here to arrive at equations which are valid for ϵ and $\tan \delta$ of oil-impregnated paper.

The Theory of the Oil-impregnated Paper Dielectric. An ideal dielectric becomes polarized in an electric field. The positively and negatively charged particles will be displaced in the direction of the electric forces. A restoring force will limit this displacement. The basic equation will be

$$m\frac{d^2s}{dt^2} + ks = -e\left(E_0 + \frac{4\pi}{3}P_0\right)e^{i\omega t} \quad \ldots \quad (5)$$

where m . . . mass of displaced particle.
s . . . displacement of particle.
t . . . time.
k . . . restoring force per unit length of displacement.
e . . . electric charge of particle.
E . . . amplitude of electric field which is oscillating at a frequency of $\omega/2\pi$.
P_0 . . . polarization.

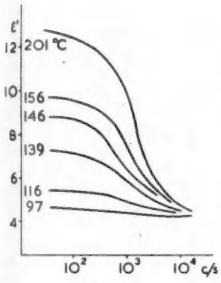


Fig. 3.

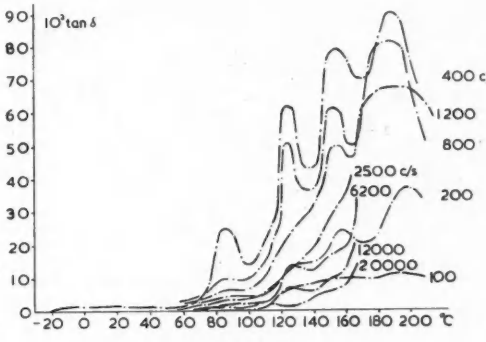


Fig. 4.

This is the simple differential equation of forced oscillations, the solution of which gives the amplitude of the displacements:

$$s = \frac{-e\left(E_0 + \frac{4\pi}{3}P_0\right)e^{i\omega t}}{m(\omega_0^2 - \omega^2)}, \text{ where } \omega_0^2 = \frac{k}{m}$$

The dielectric susceptibility is proportional to the displacement s and to the number of polarized particles in unit volume, N_∞, and we arrive at an expression for ϵ:

$$\epsilon = 1 + \frac{4\pi N_\infty e^2}{m(\omega_0^2 - \omega^2)} \quad \ldots \quad \ldots \quad (6)$$

which expresses ϵ as a function of the frequency. ϵ in this case is independent of the temperature, and the loss angle is zero.

To arrive at an expression which relates ϵ and the loss angle to temperature, we assume that the motion of the particles is damped proportionally to the velocity. The damping coefficient $\beta = A\,e^{B/T}$ decreases rapidly with increasing temperature T; A and B are constants. The modified differential equation is then:

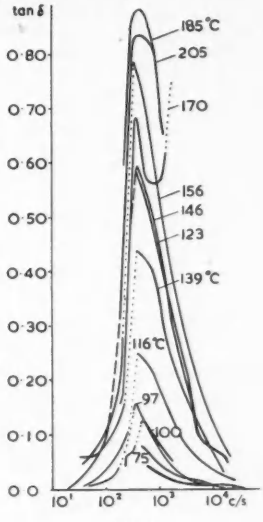


Fig. 5.

$$m\frac{d^2s}{dt^2} + \beta\frac{ds}{dt} + ks = \epsilon e E_0 e^{i\omega t} \quad \ldots \quad (7)$$

By solving this equation we arrive at expressions for ϵ and tan δ:

$$\epsilon = \epsilon' - i\epsilon'' = \epsilon_\infty\left[1 + \frac{\kappa(1-x^2)}{(1-x^2)^2 + 4\psi^2x^2} - i\frac{2\psi\kappa x}{(1-x^2)^2 + 4\psi^2x^2}\right] \quad (8)$$

where $\kappa = \dfrac{4\pi^2 N_n e}{\epsilon_\infty E_0(1+\cos\tau)}$, $\psi = \dfrac{A\,e^{B/T}}{4km}$, $x = \dfrac{\omega}{\omega_0}$

N_n . . . number of oscillating particles per unit volume.

τ . . . constant depending on the structure of the material.

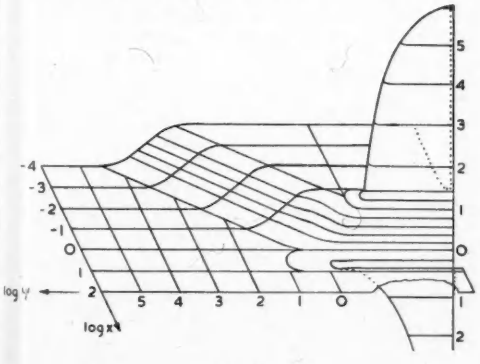


Fig. 6.

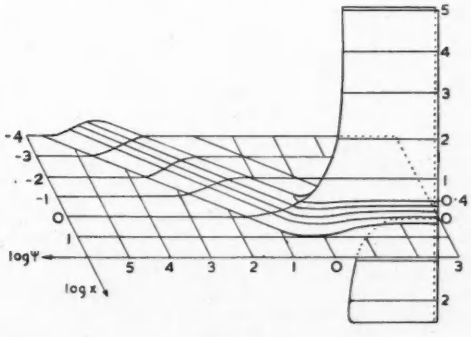


Fig. 7.

$$\tan \delta = \frac{\epsilon''}{\epsilon'} = \frac{2\,\psi\,\kappa\,x}{(1-x^2)^2 + 4\,\psi^2 x^2 + \kappa(1-x^2)} \quad (9)$$

The solution of the eq. (8) has been very successfully applied to the calculation of dispersion in the optical range of electromagnetic waves. Where the range of frequencies used for experiments on glycerin and oil-impregnated paper is concerned, it has to be assumed that the oscillating particles are comparatively very heavy.

To obtain a picture of the temperature and frequency dependence of ϵ' and tan δ, three-dimensional graphs of the expressions proportional to ϵ',

$$f(x,\ \psi) = \frac{1-x^2}{(1-x^2)^2 + 4\,\psi^2 x^2}$$

$$\tan \delta = \frac{2\,\psi\,x}{(1-x^2)^2 + 4\,\psi^2 x^2 + (1-x^2)} \quad (10)$$

have been drawn (see Figs. 6 and 7) as functions of log ψ and log x (where log ψ is the temperature-dependent axis and log x the frequency-dependent axis, and the vertical axis is $f(x,\ \psi)$, and tan δ respectively).

The figures can be compared with the measurements of ϵ' and tan δ on glycerin and oil-impregnated paper. The ϵ' and tan δ curves for glycerin in the region of log $x < -1$ will be found to present an *S*-shape and peak respectively. Oil-impregnated paper is characterized by the curves in the region of log $x > -1$. The values of ϵ' and tan δ approach maxima for the resonance frequency of the particles, $\log x = \log \frac{\omega}{\omega_o} = 0$, and high temperatures, log $\psi < 0$, where the damping is small.

These curves also have a negative part, i.e. negative dielectric constants and loss angles. To explain this difficulty, the author assumes that the oscillating particles are of varying masses, being distributed around a value according to Gauss's probability function (similar to Wagner's distribution function). He thus arrives at expressions which give no negative values for ϵ' and tan δ. In Fig. 8, $\epsilon' = f(\log \omega, \log \psi)$ is shown within the range covering technical to optical frequencies. There are three peaks of ϵ' at high temperatures, which represent the resonance region.

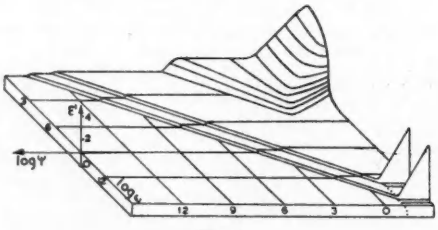


Fig. 8

Eq. (8) expresses the relation between ϵ' and the temperature, ranging from normal to very high temperatures, where thermal breakdown might occur. By measuring the properties of the dielectric at normal temperatures, the constants of eq. (8) can be found. By knowing the frequency and maximum temperature, we can then calculate whether the dielectric will be safe against thermal breakdown. Thus, we can not only avoid the testing of the dielectric subjected to very high potentials, but also avoid damage to it.

This theory might be helpful in further research on insulating materials. Figs. 6 and 7 show that there are three basic regions for a dielectric: (1) ϵ and tan δ small; (2) ϵ big, tan δ small; (3) ϵ and tan δ large. The difference of these values depends upon the size and number of oscillating particles.

CONCLUSION

The resonance theory of the dielectric outlined above originates from the theories of Maxwell, Wagner and Debye and attempts to place them on a common basis. We can draw graphs of ϵ and tan δ as functions of the temperature and frequency and are able to draw conclusions as to the possibilities of thermal breakdown.

Portable Hardness Tester for Sheet Metal

By SH. S. MANEVICH. (From *Zavodskaya Laboratoria, Russia*, No. 3, 1948, pp. 378-379, 3 illustrations.)

THE author has developed a small portable Brinell hardness tester for use on sheet metal, especially for non-ferrous metals. This device allows of making tests on sheet without the necessity of cutting off sample pieces, and thus results in a considerable saving of time and material. The accuracy of the tester varies from 2-3 Brinell units for soft metals of a hardness up to 50, to 7-10 units for comparatively hard metals having a hardness of 120-170 units. The weight of the tester is $2\frac{3}{4}$ pounds and it consists essentially of two handles operating a test head fitted with an indicator and three prongs. A sketch of the head is shown in Fig. 1, from which it will be seen that the two outer prongs have rounded ends (4 mm diameter) and the central prong is conical, the radius of the tip being 0·1 mm. The prongs are made of hard steel of 60 Rockwell C hardness. When the handles are pressed together, the springs in the head are compressed and drive the prongs into the metal. When the handles are fully compressed, the force exerted by the central spring on the pointed prong is 18 kg, while the force on the rounded prongs is 2 kg. These forces can be varied by the choice of springs. In view of the difference in the force and in the shape of prongs, the depths to which these enter the metal are different, and this difference is shown on the indicator.

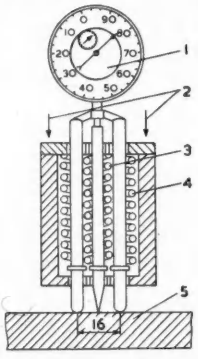


Fig. 1. Diagrammatic sketch of the head of the portable hardness tester.

(1) Indicator; (2) direction of force; (3) operating spring for the pointed prong; (4) operating spring for rounded prong; (5) test piece.

The tester must be calibrated with samples of known hardness and a calibration curve is drawn which permanently accompanies the tester.

Before each series of tests, the indicator needle must be set to zero.

The Part played by Oxygen and Nitrogen in Arc-Welding

By J. D. Fast. (From *Philips Technical Review*, Vol. 10, No. 1, July, 1948, pp. 26-34, 4 illustrations.)

(*Concluded from November issue.*)

Solubility of Nitrogen in Liquid Iron and Absorption of this Element when Welding with Bare Electrodes

Considering that welding is usually done in air, at a pressure of 1 atm, the partial pressure of nitrogen then being about 0·8 atm, the solubility at 0·8 atm nitrogen pressure is of particular importance to us.

Various investigators have determined the solubility of nitrogen in liquid iron at 1 atm nitrogen pressure. The highest and, possibly, the most reliable values were found by Kootz[8]). The solubility $C_{1\,atm}$ at 1 atm as a function of the absolute temperature T can be represented, according to his measurements, by the formula

$$\log C_{1\,atm}\ (\text{in wt. per cent N}) = -\frac{1170}{T} - 0{\cdot}715 \qquad (13)$$

Substituting for T the boiling temperature of iron, about 2750° K, we find a nitrogen content of 0·072 per cent. Considering that the nitrogen pressure is only 0·8 atm, this value has to be reduced to $0{\cdot}072 \times \sqrt{0{\cdot}8} = 0{\cdot}065$ per cent. For $T < 2750°$ K, eq. (13) would yield still smaller values of solubility.

Losana's measurements and also those of other investigators show that, when welding with bare electrodes, nitrogen may be absorbed in quantities three to four times as much as the maximum quantity (0·065 per cent, in accordance with eq. (13).

In our calculations, however, it has been assumed that the liquid iron is in contact with nitrogen of the same temperature. This is certainly not the case, for all available experimental data indicate that temperatures of at least 6000° K are reached in the welding arc. At this temperature, the concentration of atomic nitrogen in the gas phase is already about 10^5 times as great as that at the boiling point of iron.

INJURIOUS EFFECTS OF OXYGEN AND NITROGEN

Porosity

As already stated, highly porous beads are obtained when welding with bare electrodes. This is caused by the formation of CO as a result of the reaction between the oxygen absorbed while welding and the carbon always present in commercial iron and steel.

The foregoing considerations lead us to presume that nitrogen is also to be regarded as a cause of porosity. In order to investigate this, beads were welded on carbon-free plates with carbon-free rods and, as was expected, these were found to be almost as porous as the beads obtained when using normal technical wire and plates, both containing about 0·1 per cent C.

Mechanical Strength

Owing to the great porosity and the high oxygen and nitrogen content, welds made with bare electrodes have such a low mechanical strength that there is hardly any sense in measuring their impact value. Welds made with modern coated electrodes have much lower, but still relatively high, oxygen and nitrogen contents, the nitrogen content varying from about 0·005 per cent to about 0·033 per cent, according to the type of electrode and the oxygen content from about 0·03 to about 0·12 per cent. It appears that a lower nitrogen content is always accompanied by a lower oxygen content and a higher impact value. Since the oxygen and nitrogen contents vary in the same direction, it is not possible to determine from tests on welds the effect which each of these elements separately has on the impact value.

We have, therefore, carried out a number of experiments with iron having the composition of bare electrodes, first by removing oxygen and nitrogen by repeated high-frequency melting in pure argon, and then by adding known quantities of one impurity at a time (nitrogen or oxygen). These experiments have shown that up to a content of 0·033 per cent, nitrogen does not affect the impact value, whereas the addition of oxygen up to the aforementioned content of 0·12 per cent is accompanied by a gradual lowering of the impact value.

Furthermore, the experiments indicate that the great influence of oxygen on the impact value is due to the fact that this element is present in the form of an oxide which is partly contained along the crystal boundaries. Consequently, the metal readily fractures between the crystals.

Ageing

Many investigators attribute strain ageing to the presence of oxygen, whilst others ascribe it to the presence of nitrogen. The cause of the controversy lies in the fact that the experiments have been carried out with technical steels containing both oxygen and nitrogen.

By studying this phenomenon with iron and steel, not containing both these impurities together, it has been possible to arrive at the conclusion that nitrogen causes strain-ageing, whereas oxygen does not.

Apparently this is related to the fact that nitrogen has a certain solubility in alpha-iron (a solubility decreasing with temperature), whereas oxygen is practically insoluble in alpha-iron. Consequently, supersaturated solutions of nitrogen in alpha-iron may easily be obtained, for instance, by rapid cooling from P to Q in Fig. 3. In the case of great supersaturation, there may be a spontaneous precipitation of finely divided Fe_4N (quench ageing). If there is only slight supersaturation (for instance, after comparatively slow cooling), then one can imagine that mechanical working is necessary to start precipitation (strain-ageing). It is also possible that in the latter case it is not so much a question of precipitation of a new phase as the occurrence of changes in concentration in the homogeneous solution, which have an adverse effect upon ductility.

As Table III shows, the solubility of nitrogen in alpha-iron at room temperature is at most a few ten-thousandths per cent and therefore even very small quantities of nitrogen may be expected to cause ageing phenomena.

USEFUL EFFECTS OF OXYGEN AND NITROGEN

Transmission of Heat

Wyer[9] has pointed out that heat transfer to the metal is governed to a high degree by dissociation of the gas in the arc, not only when welding in hydrogen but also when welding in air.

Dissociation of the oxygen and nitrogen requires a great deal of energy and, consequently, the arc voltage is greater than that in a monatomic gas. The atoms re-combine for the greater part on the surfaces of the electrodes, thereby again releasing their dissociation energy.

Transfer of the Metal

The influence which oxygen in combination with carbon exercises upon the transfer of the metal from the electrode to the workpiece is of fundamental importance.

At first sight, it always appears strange that it should

[8]T. Kootz, Arch. Eisenhüttenwes. **15**, 77-82, 1941.

[9]R. F. Wyer, Gen. Electr. Rev. **42**, 170-172, 1939.

be possible to weld "overhead," the droplets being "shot" upwards against the force of gravity.

To throw more light upon this question, we have carried out some experiments with 4 mm wire made of carbon-free iron, and overhead welding was found to be impossible. The experiments were repeated with the same kind of iron, but with 0·1 per cent C added, when it was indeed found possible to obtain an upward transfer of the metal.

When welding with bare electrodes, the C content drops to very low values as a consequence of the reaction with oxygen. The experiments described above give the impression that this reaction leads to small explosions in the melting metal, which act as propelling forces upon the drops.

However, overhead welding with bare electrodes, even if they contain carbon, is hardly practicable on account of the fact that, owing to the absence of a "cup," much of the metal is thrown off laterally.

If the formation of CO in the metal is indeed the propelling force when welding with bare electrodes, then not only the absence of carbon (even when oxygen is present) but also the absence of oxygen (even when carbon is present) must neutralize the welding action proper. This conclusion is borne out by experiments of Doan and Smith[10]). They welded with bare electrodes in different gas atmospheres. In helium, argon and nitrogen, the droplets just fall, apparently under the action of gravity. There is no arc blow and no crater is formed (an elliptical depression in the molten or solidified pool underneath the tip of the electrode). Consequently there is little penetration. The addition of only a few per cent of oxygen to the inert gas was sufficient to make the welding normal; the droplets are then driven out with force, a crater is formed and the penetration is normal.

In welding with coated electrodes too, the development of gases is fundamental for the transfer of the metal. With the best coated electrodes, however, these gases are not formed by the reaction between oxygen and the carbon in the metal, because the coating affords protection against this oxidation. The production of gases then takes place in the coating and constitutes one of the most important functions of the coating.

[10]G. E. Doan and M. C. Smith, Welding J. **19**, 110s-116s, 1940.

A New Isolated-phase Metal-enclosed Busbar.

By B. W. Wyman and R. B. Shores. (From *Electrical Engineering, U.S.A.*, Vol. 67, No. 6, June, 1948, p. 564, 2 illustrations.)

The design of bus runs for high capacity generating stations always has received considerable attention in order to ensure freedom from faults of any type. A metal-enclosed non-ventilated isolated-phase busbar has been developed which combines a high degree of reliability with immediate accessibility for inspection or maintenance. The following specifications were established to guide the development of the new design:

1. Ultimate design is to permit shipment of complete factory-assembled sections to reduce field installation time to a minimum.

2. Designs must permit maximum accessibility for inspection or maintenance so that any insulator, or bus section, can be inspected, or even removed and replaced, in the shortest possible time. As some buses will be installed in hazardous locations, such as on scaffolding or suspended from ceilings, unbolting or handling of heavy, loose covers is to be avoided by hinging the covers.

3. The number of porcelain insulators is to be kept to a minimum consistent with strength requirements, in order to reduce the number of creepage surfaces as each creepage surface constitutes a potential flash-over point.

4. Angular and longitudinal flexibility should be designed into the bus and enclosure junctions so that no high accuracy alignment is necessary in the field which would increase the cost of installation.

5. Longitudinal flexibility is to be designed into the bus junctions to provide for thermal expansion and contraction of long runs without introducing strains in either bus supports or porcelain insulators.

6. The metal enclosures are to be dust-tight to prevent reduction of the flash-over strength over insulator surfaces.

7. The metal enclosures should be weather-tight so that they can be installed either indoors or outdoors without special construction.

8. Temperature rise limitations, 60-cycle high-potential endurance tests, and basic impulse endurance tests are to be in accordance with AIEE Standards for station-type switchgear.

9. A generous margin of mechanical strength is to be designed into the bus supports to withstand the maximum magnetic forces associated with the rated momentary short-circuit currents.

10. Metal enclosures are to be constructed of non-magnetic material to eliminate magnetic heating and thereby permit minimum cross-section conductors.

11. Seal-off bushings are to be provided at appropriate locations to act as arc and gas barriers to prevent the possibility of a phase-to-earth arc within an enclosure running along the entire bus length.

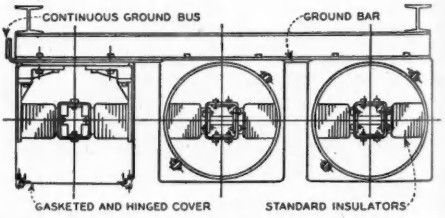


Fig. 1. **A 3-phase assembly of the new bus consisting of three individual phase housings suspended from two H-beam cross-members which constitutes a 3-phase shipping section.**

Details of the completed bus design are shown in Fig. 1. The enclosure consists essentially of a single aluminium sheet shaped to form three of the sides, while the fourth side consists of a gasketed hinged cover. The channel bus is supported on standard porcelain insulators. Silver-surfaced semi-flexible copper connections are used to connect the end of each channel section to the next section, thereby providing angular flexibility, as well as providing for thermal expansion and contraction of the bus. The porcelain insulators supporting the bus are mounted, in turn, on removable heavy steel supporting brackets which are bolted solidly through the housing to the main H-beam cross-member. Each end of the housing fits into the gasketed groove of an aluminium end plate. A circular flange with an insulating gasket around its circumference projects from these end plates. Simple split aluminium banding rings are used to span the distance between the end plates of adjacent sections. A heavy continuous longitudinal ground bus extends the full length of the bus run and is connected to each section. The new design has been given complete high-potential, impulse, heat-run, and weather tests. Sample sections also have been subjected to momentary current tests considerably in excess of their ratings with no evidence of any damage.

A Light-Alloy Dumping Wagon

By J. VALEUR. (From *Revue de l'Aluminium*, Vol. 25, No. 147, September, 1948, pp. 279-288, 10 illustrations.)

THIS article describes in detail the design, construction and testing of a light alloy wagon. The standard steel dumping wagon, which the light alloy wagon was intended to replace, carries a load per axle of 16 tons. Since the tare weight is about 10 tons, the pay load is about 20 tons. It was desired to effect the replacement without loss of strength, and in accordance with S.N.C.F. (French National Railways) standards. It was also desired to retain standard methods of construction, so that building and maintenance could be done by shops with experience in steel wagons, but without plant or labour specially adapted to light alloy work.

Heat-treated alloys were not used for the plates or sections, because they are difficult to weld and require special equipment for construction and repair. The alloy chosen instead was A-G5 with 5 per cent magnesium. Its mechanical properties in the annealed state (as received) are:

Ultimate tensile strength	19-20 tsi
Elastic limit	9-11 „
Reduction of area	26-20 per cent
Fatigue limit (100×10^6 reversals)..	7-8 tsi

For the castings, which are usually replaced if they become damaged, use was made of the heat treated alloy A-U5GT (APM), having the following properties:

Ultimate tensile strength ..	21-25 tsi
Elastic limit	15-18 „
Reduction of area	6-10 per cent
Fatigue limit (100×10^6 reversals)..	6 tsi

For certain safety members, which are also replaced if damaged, A-U4G (Duralumin) was used, which possesses properties superior to those of mild steel. Steel was retained for the tyres, axles, springs, couplings, the linings of the brake blocks, and various pins. All other parts were in light alloy.

The dimensions, dictated by the standard steel wagon, were as follows:

Length of underframe (excluding buffers)	22-ft 0-in
Wheel base	14-ft 9-in
Internal height of body ..	4-ft 8-in
Internal length of body ..	22-ft 0-in
Internal width of body ..	8-ft 5-in
Total height above rail ..	8-ft 8-in
Volume of body	880 cu ft

Underframe. The steel wagon is based upon two side sills held together by the intermediate and end sills. It was not possible to retain this structure as the side sills would have required an excessive depth of web. Four longitudinals of U-section with 10-in depth of web (as in the steel structure) were, therefore, used. The light alloy structure is clearly illustrated by Fig. 1, which also shows the dimensions of the sections (in mm).

It is difficult to calculate the stresses exactly, but

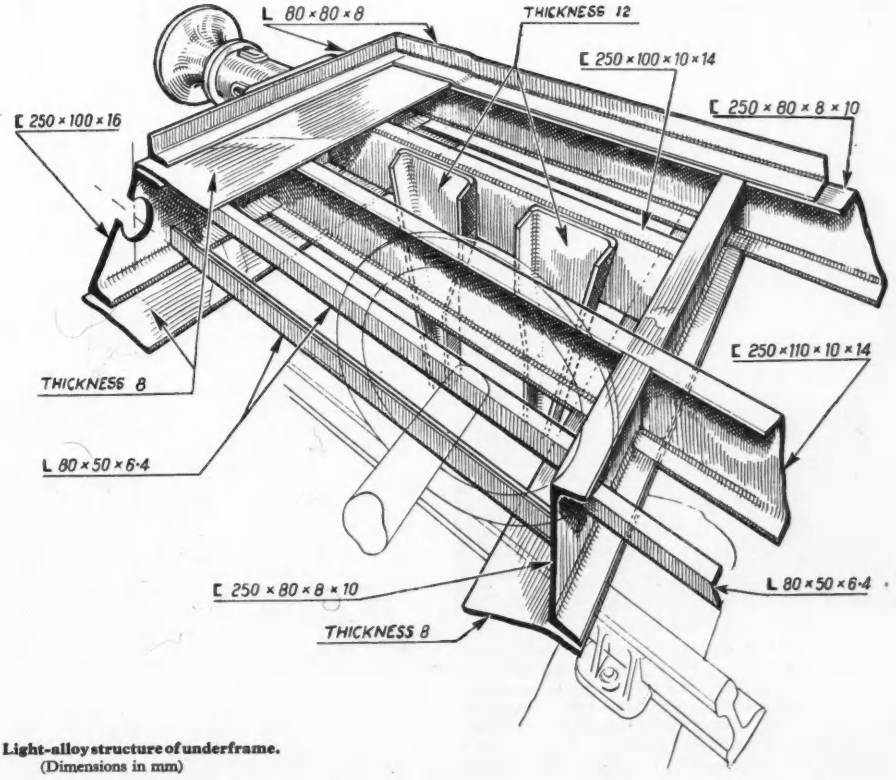
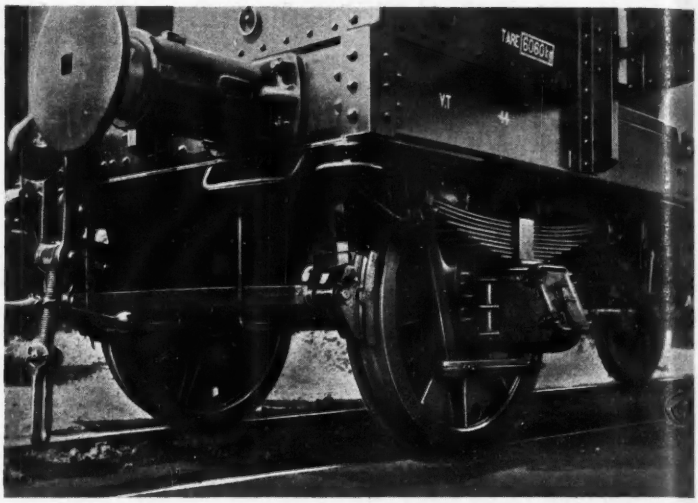


Fig. 1. **Light-alloy structure of underframe.**
(Dimensions in mm)

Fig. 2. **Axle box suspension.**



calculations were made and gave the following approximate maximum figures:

In the outer longitudinals 3·7 tsi
„ „ inner „ 4·8 „

These stresses correspond to 19 per cent and 24 per cent, respectively, of the breaking stress. In the steel wagon the maximum stress in the side sills is 5·7 tsi, say, 20 per cent of the breaking stress.

Suspension. Owing to the wide variation in bearing load between the empty and loaded conditions, it was necessary to fit a suspension of variable flexibility. Moreover, the suspension had still to be interchangeable with the normal springs in service with the S.N.C.F. The solution was a spring with nine leaves, in which the first group of three was suitably spaced from the remaining six, (see Fig. 2). This gave a deflection varying between 0·63 and 0·26 inch per ton.

One of the conditions to be satisfied was that the force required to drive the axle into the wheel (stripped of its tyre) should not be less than 30 tons. This was achieved despite the low elastic modulus of the light alloy, and without making the hub excessively thick, by adopting a continuous disk. To spread the stress increase over the hub, the disk was given an undulating section, (see Fig. 3). A further advantage of this section is its high resistance to the bending moment set up in the disk by shrinking on the tyre.

Fig. 3. **Wheel centre.**

Construction. The underframe and body were rivetted with steel rivets. Owing to the high thermal conductivity of light alloy, the rivets had to be raised to a higher temperature than usual. The faces to be rivetted were painted with bituminous protective paint.

The lining of the main bearing was done without difficulty in the S.N.C.F. shops, after tinning the light alloy by rubbing it with a stick of soft solder. The S.N.C.F. are dropping this method in favour of bronze bushes incorporated in the light alloy casting and subsequently lined.

Component Testing. Before making the wheel centres, preliminary tests were made on sleeves of steel and light alloy. These showed that:

1. With the same greasing, light alloy requires about twice as much interference as steel in order to give the same pressure of fit.
2. Large driving-on forces can only be obtained with interferences which produce a permanent stretch. After pulling off, the interference may be reduced by 30-40 per cent in the case of steel.
3. The driving-on force depends critically on lubrication. If driving-on is done after carefully wiping off the grease, the force is 50 per cent greater than after leaving the grease on.

It was finally decided to use an interference of 0·024-in to make certain of satisfying the conditions. The force obtained was amply sufficient, being over 200 tons.

Results. The completed wagon had a tare weight, in running order, of 6 tons, made up as shown in the accompanying table. The corresponding steel wagon has a tare weight of 9 tons 9 cwt. The weight saved was, thus, 3 tons 9 cwt., thanks to 2 tons 11 cwt. of light alloy.

The prototype wagon was put into service and has run successfully for nearly four years. Minor modifications, and repairs after an accident, have been carried out without undue difficulty.

WEIGHT ANALYSIS IN LBS.

Part	LIGHT ALLOY				STEEL, ETC.	WOOD
	Sections	Plates	Castings	Total		
Underframe	1580	382	0	1962	177	0
Suspension	0	182	864	1046	4830	0
Buffers and Couplings	0	0	334	334	630	151
Body	679	791	150	1620	279	1358
Brake-gear	0	336	406	742	363	0
Totals	2259	1691	1754	5704	6279	1409

GRAND TOTAL :—13,392 lbs. or 5 tons 19 cwt. 64 lbs.

Influence of Sheet or Plate Thickness on Butt Weld Bend Tests

By N. LUDWIG. (From *Werkstatt und Betrieb*, Vol. 81, No. 7, July, 1948, pp. 181-183, 2 illustrations.)

THE folding test is the cheapest and simplest deformation test for welded joints. Its result, the folding angle at the inception of cracking or fracture, is influenced by many factors which must be considered when a minimum angle is laid down in welding specifications or terms of contract. The influence of the thickness of the sheet or test piece has not yet been thoroughly investigated. Test pieces of plain carbon steel of 37 tons per square inch tensile strength, with traces of copper and chromium, were, therefore, welded under factory conditions, and folding tests were carried out on pieces of 6, 12, 15, 25, 30, 40, 45, and 50 mm ($\frac{1}{4}$ to 2 in) thickness. (Table I.)

The difficulty of deformation tests on welded test pieces consists in obtaining the maximum deformation within the weld seam. Several test methods have been developed but none of them showed any marked advantage over the simple folding test for sheet thicknesses of more than 5 mm ($\frac{3}{16}$ in)[1]. With sheets of less than 5 mm thickness, the influence of the weld on section modulus and grain structure is so strong that the maximum deformation of test pieces always occurs outside the region of the welded seam[2]. In the usual folding test, the test piece is supported on rollers and bent round a mandrel until the first clearly visible crack appears. The final angle of bending is a measure of the deformation. Other measures have been suggested but are apt to lead to difficulties[2].

The influences on the folding angle of (a) distance between supports; (b) diameter of support rollers; (c) diameter of mandrel; (d) surface, as welded or machined flat; (e) welding process; (f) tensile strength of parent and weld metal have been satisfactorily elucidated[3, 4]. The influence of the bending speed has not been investigated, but can be deduced from the fact that ductility decreases with speed of deformation. The importance of the width and thickness of the test piece and the angle of plate chamfer have not yet been clearly established[4, 5] although some investigators[5] assume that the folding angle decreases with increasing plate thickness. In order clearly to establish this relationship, the following tests were conducted on one type of steel only (see Table I), and with one welding process.

TABLE I. CHEMICAL ANALYSIS AND TENSILE STRENGTH OF PLATE MATERIALS.

Plate thickness mm	Constituents, per cent								Tensile strength kg/mm²
	C	Mn	Si	Cu	Cr	Mo	P	S	
6	0·18	1·27	0·46	0·09	0·05	<0·01	0·049	0·027	65
12	0·18	1·03	0·44	0·39	0·10	<0·01	0·032	0·018	58
15	0·18	1·27	0·37	0·25	0·09	0·04	0·058	0·020	59
25	0·19	1·00	0·25	0·44	0·33	0·04	0·036	0·012	62
30	0·20	1·27	0·52	0·35	0·09	0·04	0·029	0·012	63
40	0·19	1·16	0·31	0·45	0·43	<0·01	0·032	0·026	65
45	0·19	1·27	0·41	0·20	0·09	0·04	0·035	0·031	61
50	0·19	1·27	0·41	0·20	0·09	0·04	0·035	0·031	61

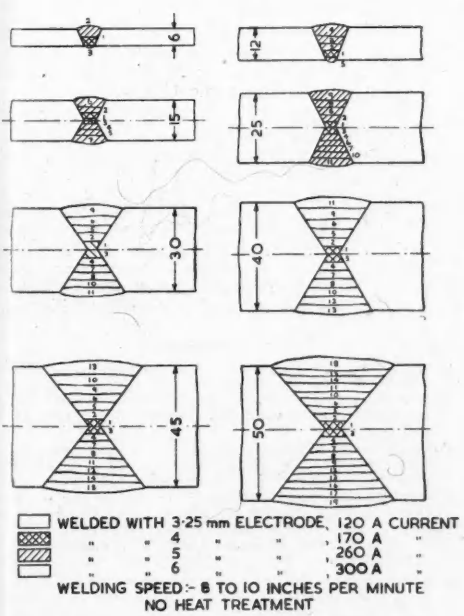


Fig. 1. **Welding procedure.**

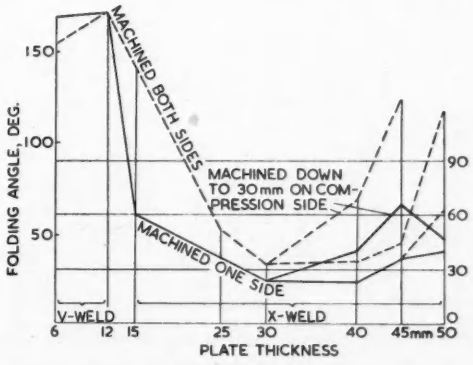


Fig. 2. **Folding angle as a function of the plate thickness.**

Two pieces 16 × 6 inches were cut from every sheet and welded to form plates of 16 × 12 inches. In addition, pieces 16 × 10 inches were cut from the three thickest plates and welded to form plates of 16 × 24 inches. Shielded electrodes were used throughout (Class E 52z, German Standard DIN 1913), and the test pieces were welded by electric arc in a factory workshop, in accordance with Fig. 1. Nine folding test pieces of 30 mm ($1\frac{3}{16}$ in) width were cut from each welded plate, six of these were machined flat on one side only, three were machined on both sides to nominal plate thickness and 5 mm graduating lines were inscribed over a width of 50 mm (2 in) on the tension side of the test piece. All edges were rounded. One

TABLE II. FOLDING ANGLES.

Plate thickness mm	No. of machined sides	Folding angles		Plate thickness mm	No. of machined sides	Folding Angles	
		results[2] deg.	mean deg.			results[2] deg.	mean deg.
6	1	180 wc 180 wc 180 wc 180 wc 180 115	169	40	1	29 23 22 22 20 11	21
	2	180 180 99	153		2	40 40 22	34
12	1	180 wc 180 wc 180 wc 180 wc 180 139 f	173	45 (30)[1]	1	134 f 62 58 51 47 30	64
	2	180 wc 180 wc 152	174		2	142 136 94	124
15	1	69 66 60 58 56 56	61	45	1	43 41 36 35 22 —	35
	2	180 wc 180 wc 73 —	144		2	51 43 44 39	44
25	1	49 42 42 33 23 16	34	50 (30)[1]	1	66 33 34	45[3]
	2	62 46 42	50		2	—	—
30	1	32 32 30 25 11 7	23	50	1	(143) 53 50 41 21 —	62
	2	31 31 30	31		2	143 140 139 46	117
40 (30)[1]	1	52 50 48 40 32 16	39		—	—	—
	2	74f 73 55	67				

[1]Machined down to 30 mm on compression side.
[2]Where no letters are shown a crack appeared; wc = without crack; f = fracture.
[3]Six specimens fractured owing to fault in parent plate.

half of the test pieces of the three thickest plates (40, 45 and 50 mm), were machined down to 30 mm on the compression side (German Standard DIN V 50 121). The test equipment consisted of support rollers of 100 mm (4 in) diameter, the diameter of the bending mandrel was always three times the plate thickness, and the opening between support rollers was set to six times the plate thickness of the test piece. Test pieces which did not show any crack after 120 degrees bending were folded by pressure on both arms until a crack appeared, or up to an angle of 180 degrees. The mandrel movement was limited to a speed of less than 2½ inches per minute.

Results are given in Table II and the mean folding angles represented on the graph Fig. 2. The folding angle is a minimum for plates of 30 and 40 mm (1 3/16 and 1 5/16 in) thickness. The sudden decrease of the folding angle from 12 to 15 mm (½ to 1⅝ in) plate thickness may be due to the change-over from V- to X-weld. The increase of angle for plates thicker than 40 mm (1 5/16 in) may be accidental or due to the fact that the greater number of layers exerts a relatively greater annealing effect than the smaller number of layers on the thinner plates. Chemical analysis or tensile strength of the respective plates does not explain this behaviour. Test-pieces machined flat on both sides show a greater deformation than those machined on one side only, a result contrary to that obtained by other investigators working with bare electrodes[4]. Thick test-pieces which were machined down to 30 mm thickness show a definitely improved deformation, another indication that the V-weld gives more favourable results in folding tests than the X-weld. The elongation of the extreme tension fibre was also measured on specimens machined on both sides. Except in two cases, the elongation at the appearance of the first crack was never below 20 per cent. No clear relationship between these elongations and the corresponding folding angles could be established.

Judging from the result of these tests, it is proposed that in Welding Specifications, minimum requirements regarding the folding angle should be graded according to the plate thickness. For a mandrel diameter of three times the plate thickness, this grading should be:

Sheets up to 12 mm (½ in), minimum folding angle 90 deg.

Over 12, up to 25 mm (1 in) minimum folding angle 40 deg.

Over 25 mm, minimum folding angle 20 deg.

Moreover, a minimum elongation of the extreme tension fibre of 20 per cent on the appearance of the first crack could be laid down. Further elucidating tests on mild steel and 32-ton steel plates are suggested. The material should come from one charge, part of the test-pieces should be annealed, and 10 mm and 15 mm plates should be welded with both a V-weld and X-weld.

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DESIGN AND WORKSHOP

A Digest of articles of interest to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers, and all those concerned with the practical aspects of Engineering.

Designing for Silver Brazing by Induction Heating

By F. W. CURTIS. (From *Product Engineering, U.S.A.*, Vol. 19, No. 8, August, 1948, pp. 109-113, 9 illustrations).

WITH a suitable design, often silver brazing by induction heating is the lowest cost way for joining metal parts into assemblies.

In silver brazing, as the work is heated the silver alloy melts, and flows throughout the joint by capillary action. Induction brazed joints: (1) Can be made exceptionally strong; (2) are liquid and gas tight; (3) withstand vibration and shock; (4) are unaffected by normal temperature changes; (5) provide good electrical conductivity; and (6) can be plated conventionally.

The strength of a silver-brazed joint depends upon the area and shape of the contact, and the clearance or the accuracy of the fit between the parts to be joined. As in any type of adhesive joint, butt assemblies or plain tension fits should be avoided. Very often the difference between success and failure will depend on this consideration.

SOME DESIGN POINTERS

In Fig. 1, the assembly (*A*) represents a poor design. Not only is the joint weak, but alignment of the two parts would present some difficulties during heating. Joint (*B*), with a slight counterbore for the pin, is much stronger. Design (*C*) is good and ensures maximum strength. (*D*) is excellent because heat can be localized to the projecting hub area. Another poor design (*E*) has no shear area. By inserting the disk inside, as in (*E*), the strength of the joint is far better. Designs (*G*) or (*H*) are best, since alignment of both parts is definitely assured. In fitting a screw-machine insert to a tube, design (*J*) incorporates strength; but unless located by a fixture, end location is not positive and the parts may shift. Designs (*K*), (*L*), and (M) are better because end location is assured and alignment in a heating coil is simple.

Usually, an overall clearance of from 0·0015 to 0·003-in is allowed for small diameters, and 0·002 to 0·006-in for larger assemblies. Press fits, where a definite interference exists, are not practical because no space remains for the silver alloy to fill. But with excessive clearances between the parts to be joined, the strength of the joint falls off to that of the brazing alloy. With proper fit, however, the joint can be made as strong as the material being brazed.

Silver alloy can be applied in the form of wire, thin sheet, or powder. Wire is the most common. In Fig. 2, wire is applied by hand in (*A*) while the work is revolved slowly as it is heated. The wire ring method, (*B*) and (*C*), is preferable because uniform distribution of alloy is assured. Wire rings and inserts often are formed to various shapes by coiling the wire on a mandrel and then cutting through them lengthwise. Thin sheets, or shim alloy can be blanked into washers or inserts as shown in (*D*). Wire sizes mostly run from $^1/_{64}$ to $^1/_{16}$-in, and shims are made of 0·002, 0·003, 0·005-in thick ribbon stock. Wire rings are less ex-

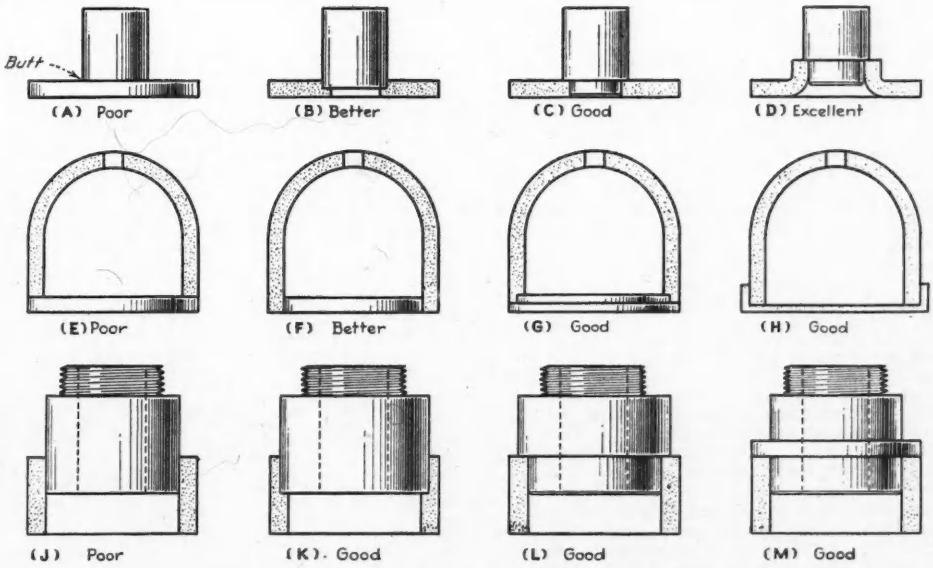


Fig. 1. Comparison of typical designs for assembly by silver brazing.

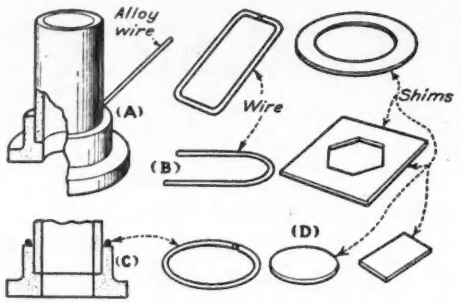


Fig. 2. **Wire and shim shapes for applying silver brazing alloy.**

pensive than shim inserts. Use of granulated silver brazing alloy, or powder, is limited to specific applications.

Different ways of placing wire rings on the work are shown in Fig. 3. In (*A*), when the shaded area is

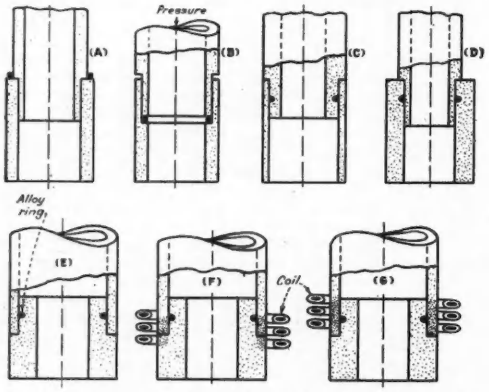


Fig. 3. **Proper positioning of silver alloy ring and induction heating coil is essential for best results.**

heated, the alloy will flow throughout the shear joint. With the ring placed inside as in (*B*), pressure should be applied to the upper tube as the alloy flows. Far better from a design standpoint is to insert the alloy wire ring into a groove, either machined in the insert, as in (*C*), or inside the sleeve as at (*D*). When the assembly is heated to the melting point of the silver, the alloy leaves the groove and flows to the shear area.

When heated to their flow point, silver brazing alloys are extremely fluid and tend to flow more in the direction from which the heat is applied. With the assembly (*E*), the alloy in the inserted ring will flow more towards the outside edge of the joint when the coil is positioned as shown in (*F*). If the coil is raised relative to the area to be brazed to concentrate heat above the alloy ring, the silver will first flow upward and form a fillet on the inside surface, and just show through at the outside edge. This control of flow is often advantageous when excessive alloy is not wanted on inside finished surfaces.

Alloys for silver brazing are made in various compositions having melting points from 1,000 to 1,400° F. A typical alloy containing 50 per cent silver with cadmium and copper melts at 1,150° F and flows at about 1,200° F.

In certain designs, inserts can be brazed to heat treated parts without affecting the hardness. Conversely, often it is possible to harden a localized area after joining without affecting the braze.

As in the joining of any metal parts, all surfaces to be brazed must be clean before heat is applied. Flux is necessary for proper flow of the alloy to the surfaces to be joined.

INDUCTION HEATING

Induction heating for silver brazing usually offers good heat control and a high degree of accuracy. Often advantages include:

1. Economical operation, especially where only a local surface is to be heated.
2. Heating is extremely fast so that increased output is gained.
3. Uniform heating is ensured once a time cycle has been established because of the timing controls, thus reducing spoilage to a minimum.

Heating coils for silver brazing can be designed in a variety of ways, such as shown in Fig. 4. Single-turn coils are made of tubing in (*A*), or of solid bus construction as in (*B*). Also copper tubing can be used for multi-turn coils (*C*), or for multiple-position coils (*D*). A multiple-position single-turn coil, or solid inductor, is shown in (*E*). A feed-through conveyor-type coil (*F*) is commonly used for the brazing parts progressively.

Where the band of heat is narrow in relation to the diameter, and where the material is of a ferrous type such as steel, single turn coils are used. Where the band of heat is wider, as for non-ferrous materials because of the better electrical loading characteristics, multi-turn coils are used. For ferrous materials, the coupling or distance from the work being heated to the inside surface of the coil, is somewhat "looser" than would be used for hardening where a rapid transfer of energy is required. For non-ferrous materials, the coupling is much closer, usually from $^1/_{16}$ to $^1/_8$-in.

Induction heating can braze two or more surfaces of an assembly at the same time, but there are certain limitations. First, it must be possible to safeguard against the shifting of the silver alloy out of position. Second, the wall thicknesses should be somewhat alike to prevent over-heating a light joint while simultaneously heating a much heavier mass. To some extent heat compensations can be made by difference in coil couplings ; but with too wide a difference separate heating operations are to be preferred.

SOME DESIGNS COMPARED

The metal assemblies shown in Fig. 5 are representative of designs using silver solder and induction heat. The assembly (*A*) includes a steel shell, a reinforced disk and a bushing. The silver-alloy wire ring, around the head of the bushing, will flow around the

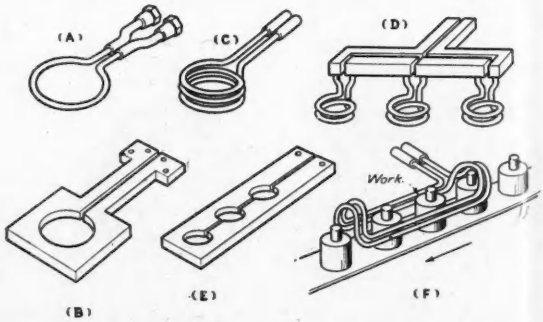


Fig. 4. **Different types of induction heating coils.**

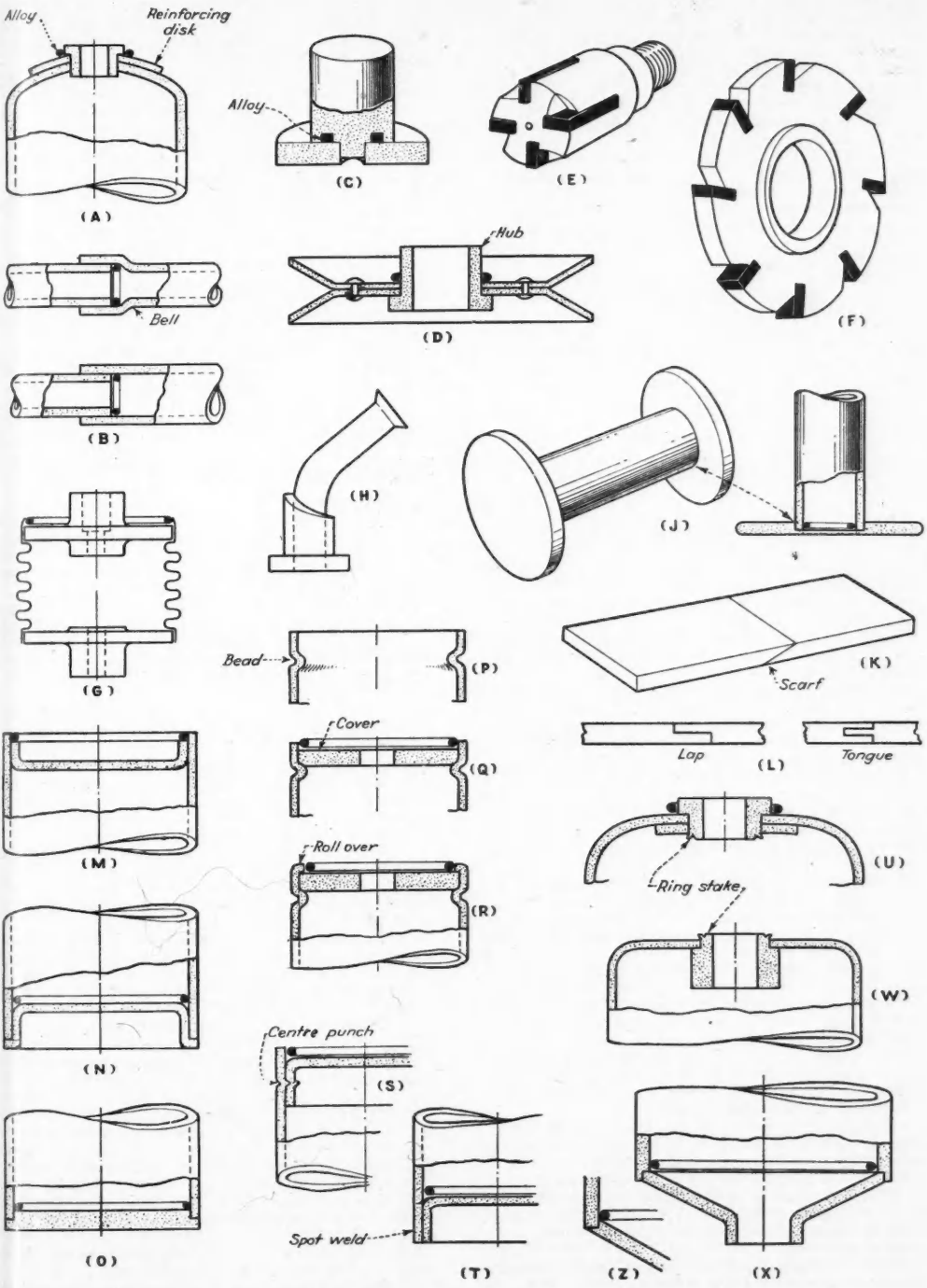


Fig. 5. Wide variety of good designs showing range of metal assemblies which lend themselves to silver brazing by induction heating.

joint faces to form a strong bond. The heating coil for such an assembly is usually a multi-turn pancake of the inside type. Heating from the underneath side causes the alloy to be drawn in from the top to the shear areas. In (*B*) are shown two forms of tube joints; the upper is the bell-type, and the lower is the straight sleeve type. Assemblies of this kind are best handled in a vertical position to ensure even distribution of the alloy.

Example (*C*) shows a method for inserting the alloy ring into a machined groove in the face of the pin. For the sheet-metal pulley with hub (*D*) a single-turn coil placed around the head of the hub ensures that the alloy will be drawn through all contacting surfaces. Mechanical parts to which carbide inserts are joined, such as carbide cutters, can be processed by silver brazing. When the inserts are arranged, such as with the reamer (*E*), they can be brazed simultaneously by a multi-turn coil. For an assembly such as the cutter (*F*), each blade is brazed separately by a formed hairpin coil. The ability to localize induction heat reduces the danger of overheating adjacent areas, or applying sufficient heat to affect a previously brazed insert.

Many varieties of the bellows assembly (*G*) and tube elbow (*H*) can be brazed in multiple set-ups. For the spool (*J*), a wire ring insert is placed into the groove of the flange, and a pancake-type coil is used for heating.

The scarf joint (*K*) is superior to a straight butt joint for joining flat parts. The lap or tongue joint (*L*) increases the mechanical strength. The assembly (*M*) is a conventional method of joining a cover to a container. For higher pressure (*N*) is better; for an air-tight low-pressure joint (*O*) is satisfactory. Covers often are brazed to containers by rolling a bead as in (*P*), then brazing the cover as shown at (*Q*). For additional strength the end of the shell can be rolled over as at (*R*). The two parts should be held together during brazing, and this can be done by first centre punching indentations at several points as at (*S*), or spot welding at intervals as at (*T*).

Typical brazed assemblies (*U*) and (*W*) are joined and held together by ring staking; this is done before the heating operation. The assembly (*X*) comprises a tube and a drawn cap, for which the alloy ring is placed on the inside and the heating coil on the outside; (*Z*) shows an alternative joint for this assembly.

Copper-Aluminium Joints and Combination Materials made by Upset Welding

By R. T. GILLETTE. (From *Materials & Methods, U.S.A.*, Vol. 28, No. 1, July, 1948, pp. 70-73, 6 illustrations.)

THE development of upset welding copper to aluminium was started on a manually operated butt welder, and after a few trials a ductile weld was produced which could be bent back double at the joint. This gave enough encouragement to continue the investigation, and after a considerable time the following factors were determined.

It is well known that alloys of copper with more than a few per cent of aluminium become excessively hard and brittle, and that alloys of aluminium with more than a few per cent of copper are likewise hard and brittle.

Interpreted into weld joint efficiency, this meant that a joint had to be produced either with no alloy or with a microscopically thin layer of alloy with low concentration of one metal in the other. It was found that such a joint could be produced by the following procedure.

The samples were prepared with a square end by machining. They were then clamped in the welder with approximately 1-in overhand of the metal between the clamps. Just enough pressure was then applied to prevent arcing when the current was applied. The current was applied for a sufficient time to melt the metal at the interfaces still retaining low pressure. When the proper temperature was reached, the pressure was suddenly increased and the current switched off at the same time.

The quick increase in pressure almost completely extrudes the alloy from the joint, leaving a sharp line of demarcation between the copper and aluminium, and producing a ductile weld.

If high upset pressures are used for the entire time of making a weld, as the aluminium starts to soften due to its heating, it will upset like a flat head rivet before it becomes hot enough to weld at all.

If low pressure is used during the entire welding cycle, the copper aluminium alloy will not be extruded from the joint, and the resultant weld will be too brittle to be of any value.

After determining these factors on a manually operated machine, a fully automatic welder was designed and built. This machine consists of a modified cam-operated butt welder. Both platens are arranged to move, the right-hand one being cushioned with an air cylinder.

To operate the welder (Fig. 1), the pressure is adjusted on the air cylinder at the right-hand end of the welder to give a pressure several times higher than the coil spring shown on the left-hand platen. The motion of the cam is transmitted through the coil spring to the left-hand platen.

Both of these pressure devices are adjustable for the particular size of material being welded. The welder operates as follows:—The work is clamped and the machine started. The welding current is turned on and off, at the proper time, by limit switches and auxiliary cams on the main cam shaft which control ignitron contactors. As the metal in the joint melts, low pressure is maintained through the spring.

When the proper temperature is reached through correctly adjusted time and current values, the cam has travelled to a point where a steep rise on the cam compresses the coil spring against a stop. The left-hand platen then becomes a solid member, and the weld is upset or forged against the high cushioned pressure of the air cylinder, the welding current being turned off at the start of the high pressure application.

If the weld were to be upset against a solid stop, cracking would result in the immediate vicinity of the weld because of the hot-short condition of the metal at temperatures below the melting point.

In making this weld, heat balance is another factor which must be considered. As aluminium melts at approximately 1210° F and copper at 1981° F, and the electrical conductivity of aluminium is about 61 per cent that of copper, and as the same welding current must be used to heat both, some compensation must be made to prevent the aluminium from melting and running from the joint before the copper is hot enough to weld. This is accomplished by having about three times as much copper as aluminium protruding from the welding clamps.

Tests made to prove the reliability of the joint indicate satisfactory toughness and ductility.

From corrosion tests it appears that the welded joints were practically the equivalent of the straight aluminium and that subjecting them to salt spray or

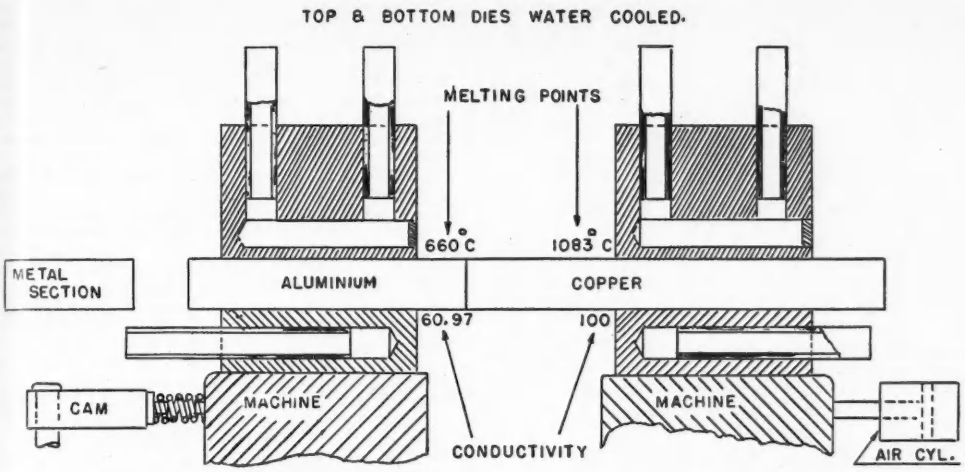


Fig. 1. **The difference in melting points of the two metals is compensated by having about 3 times as much copper as aluminium protrude from the welding clamps.**

steam did not lower their properties. The joints were also tested for their electrical resistance. No joint resistance was detected with the samples tested.

The joints also behave satisfactorily under vibration, but due to the abrupt transition from the hard copper to the soft aluminium, they are not too satisfactory under severe shock.

There is one caution to be observed in any subsequent work or fabrication in close proximity to these joints, such as welding, brazing or heat treating procedures. That is not to bring the copper-aluminium joint above the melting point of the copper-aluminium eutectic alloy, about 1018° F, as copper-aluminium in such intimate contact will rapidly diffuse into each other and form a layer of eutectic alloy which will melt at the eutectic alloy melting point, and thus the joint fails.

This joint was primarily developed to further the substitution of aluminium for copper coils on heavy, high-speed rotating electrical equipment. The use of aluminium reduces the weight of the coils by approximately 50 per cent, which is of major importance in lowering centrifugal stresses.

This type of joint is also used on the moving coil of an electrical measuring instrument wound with aluminium wire 0·005-in dia., replacing a copper winding. The reduction in weight reduces inertia so that the needle will come to rest with a minimum of swing.

The method of welding described has been used for several other applications entirely differing from coil weight reduction, where it is desirable to use aluminium to replace copper for various reasons, but where a partial use of copper is required for ease of fabrication.

Nitriding

By H. WIEGAND. (From *Revue de Métallurgie, France*, Vol. 45, Nos. 3 and 4, March-April, 1948, pp. 105-117, 17 illustrations.)

THE primary advantage of the nitriding process is the increase in surface hardness, without deformation, owing to the fact that rapid cooling is not necessary after nitriding. Nitrogen in its atomic state, between 450° and 600° C, diffuses rapidly into the steel, forming nitrides with the constituent elements, namely, aluminium, chromium, molybdenum and vanadium, these nitrides resulting in an increase in the inter-atomic attraction and thus causing an increase in the surface hardness.

Iron nitrides are also formed on the surface of the steel at higher temperatures. They are very brittle and non-malleable and may have a thickness of a few ten-thousandths of an inch. To avoid this, the nitriding temperature is kept down to 480° to 520° C, with a dissociation of only 10 to 25 per cent of the ammonia gas. Although a higher temperature would give a more complete dissociation and so be cheaper, this would entail the risk of formation of harmful iron nitrides.

A higher temperature would also reduce the time required to obtain a required depth of case, but the surface hardness would then be lower. Dual nitriding is suggested for achieving reduced time without reduced hardness, that is, nitriding for a few hours at 510° C to ensure surface hardness, followed by a period of nitriding at a higher temperature, say, 540° C.

The speed of diffusion of the nitrogen into the steel can be increased by using an electric furnace with an ionized atmosphere. The nitriding parts act as cathode, so that the temperature does not exceed 480-520° C. The nitriding time is only one-third of the normal period to give the same depth of case, but more electric power is required.

The maximum nitrided hardness is just below the surface, owing to the presence of iron nitrides in the outer surface, and the formation of carbides is kept low by keeping the carbon content of the nitriding steels low. Steels must be in the hardened and tempered condition. Nitriding gives them a long soaking at low temperature.

Diffusion of nitrogen produces an increase in volume of the outer zone which leads to changes in internal tension, as shown on Fig. 1, with compression

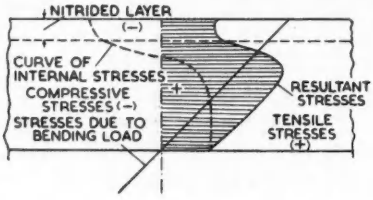


Fig. 1. **Distribution of stresses in a nitrided component under bending load.**

stresses on the outside and tensile stresses inside. The maximum stress set up by outside pressure comes just below the nitrided case, and it is at this point that fracture occurs (Fig. 2.)

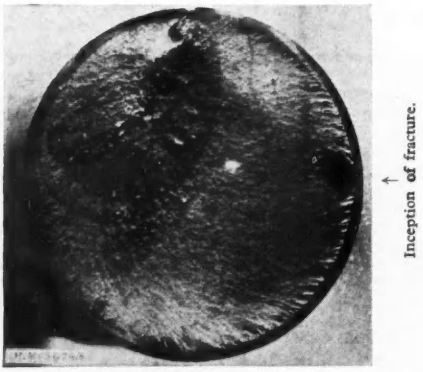


Fig. 2. **Fatigue fracture of a nitrided shaft.**

Nitrided steels resist wear, corrosion by water and other media, and fatigue. The compressive stresses in the nitrided case lead to an increase in fatigue resistance, as shown in Table I. This increase rises with the depth of the nitrided layer to a certain maximum for each category of steel, so that there is an optimum ratio (found by trial) between the thickness of the nitrided layer and the cross-section of the work-piece in question. It may not always be possible to attain this optimum, as the nitriding time would be too long. There is no relation between surface hardness after nitriding and fatigue resistance, as the latter depends on the ratio of nitride thickness to cross-section and tensile strength of the core steel. For a part subjected only to alternating stresses, a high surface hardness is not essential.

Nitrided steels are more sensitive to overloading than are non-nitrided steels.

Nitriding gives considerable protection from fretting corrosion, under alternating stress. Tests have shown that a non-nitrided steel gave a reduced fatigue resistance of $\pm$ 26 kg/mm² (17 tsi), whereas the same steel nitrided gave 55 kg/cm² (35 tsi). These results were irrespective of the material used for making the test jaws (steel, bronze or light alloy). Adequate nitriding of certain components thus allows of increased strength in the machine.

Surface roughness is of no consequence with nitrided steel, although the surface finish is so important with non-nitrided steels. The fatigue resistance of a nitrided steel did not alter, no matter whether the surface was ground, machined or emery-papered, whereas a non-nitrided steel showed a drop from 63 to 45 kg/cm² (40 to 29 tsi).

For crankshafts, the greatest increase in endurance is obtained by nitriding all over. If only part-nitriding is done, the radius as well as the pins should be nitrided. Tests have shown that radial engine crankshafts nitrided all over showed a fatigue resistance 15 per cent greater than similar crankshafts which were nitrided on the pins only. Machining of crankshafts to be nitrided all over is less costly than in the case of non-nitrided crankshafts, as the latter must have a very high surface finish to avoid the setting up of fatigue cracks.

Similar remarks apply to connecting rods, for which nitrided steel helps to overcome trouble due to fretting corrosion between the bronze bearing and the big end of the rod. A nitrided steel connecting rod under alternating stress tests stood up to 1,560,000 stress reversals as against 39,000 with the non-nitrided steel. It is, of course, of the utmost importance to avoid shock with the nitrided parts.

Nitriding is a heat treatment which allows of improvement being made in the mechanical properties of steels or of machine parts. Its essential advantage is the increase in surface hardness, which is higher than can be obtained by any other method of surface treatment. The second advantage is not so generally known. This advantage is the influence of nitriding on the internal stresses of steels and of machine parts. Applied to components subject to certain working conditions and to appreciable stresses, nitriding allows of an increase in endurance and also in the strength of the whole machine.

TABLE I. MODULUS OF ELASTICITY AND TENSILE STRENGTH OF NITRIDED LAYERS.

Test piece	Heat treatment.	Tensile Strength		Modulus of elasticity psi
		Commencement of cracks in nitrided layers tsi	Fracture of test piece tsi	
Flat bar 5/64-in × 15/32-in	Hardened, tempered, not nitrided	—	—	30,250,000 to 30,700,000
Flat bar 5/64-in × 15/32-in	Nitrided to a depth of 0·032-in	60	60	30,250,000 to 30,700,000
Round bar 3/32-in dia.	Nitrided to a depth of 0·043-in	58·5	58·5	30,000,000 to 30,700,000
Round bar 13/32-in dia.	Nitrided to a depth of 0·008-in	65	74	30,000,000 to 30,700,000
	Nitrided to a depth of 0·012-in	62	70·5	
	Nitrided to a depth of 0·018-in	60	70·5	

Chemical composition of steel :—C = 0·3 per cent ; Cr = 2·5 per cent ; Mo = 0·3 per cent ; V = 0·25 per cent.

Automatic Punching of Stator and Rotor Laminations

A. Y. Miller. (From *Vestnik Electropromyshlennosti, Russia*, No. 5, 1948, pp. 14-17, 5 illustrations.)

The punching of laminations forms a considerable part of the work involved in the manufacture of standard-type induction motors. The accuracy of punching is a factor which can greatly influence the amount of work put in, as the subsequent de-burring and assembly of laminations into packets can be reduced to a minimum when the accuracy is good.

With manual presses the checking and stacking of laminations take approximately the same time as the punching itself.

A higher efficiency in manufacture is most important with machines rated at 1 to 10 kW, which are normally mass-produced. The presses used for these ratings usually have dies of the follow-on or compound type. In the former case the slots and the inner and outer contours are punched out progressively in individual operations, while with the latter the punching is completed in a single operation. The output of the follow-on type press is normally higher, because stator and rotor laminations can be dealt with by a single die. Other advantages of this type are that laminations can be allowed to fall through the bed platen, avoiding the necessity of mechanisms for their removal, which are often complicated. The dies are more substantial, their thermal treatment, therefore, being simpler and their useful life longer.

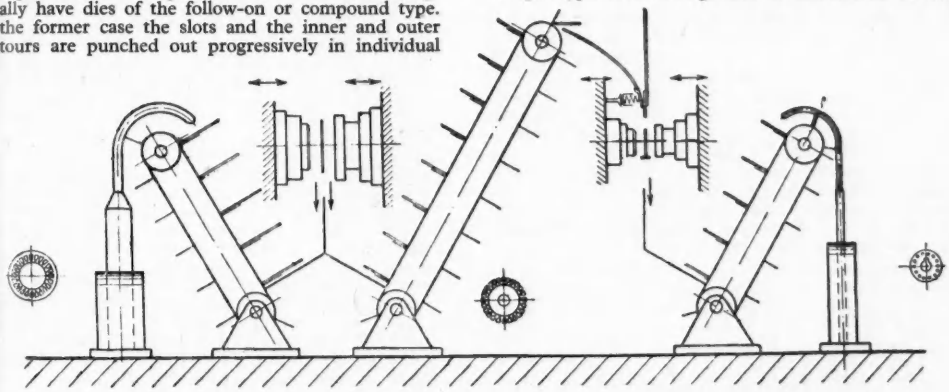


Fig. 1. **Diagrammatic arrangement for automatic punching of stator and rotor laminations on a horizontal press with twin dies equipped with automatic feed and stacking device.**

The disadvantages are as follows:—

(1) In view of the number of consecutive operations, the feed may be inaccurate, causing a degree of eccentricity of the laminations.

(2) For the same size of laminations the dimensions of the stamping die are larger than in the case of the compound press. This limits the size of punchings which can be satisfactorily obtained by this method to 700-800 mm diameter.

(3) Any irregularities of the sheet material have a marked effect on the quality of the finished laminations.

Mechanization of both types of presses can be carried out and leads to good results, but it is obtained more simply on the follow-on type press, the principal requirement being an automatic feed arrangement.

An example of fully automatic operation is shown diagrammatically in Fig. 1. The machine consists of two horizontal dies, one for stamping stator laminations and cutting out the centre, this being transported by a conveyor to the second horizontal die which finishes the rotor laminations. The main advantage of this arrangement is that the laminations can be very simply removed from the two dies.

There are certain disadvantages connected with this arrangement, such as the comparatively complicated design and an unavoidable eccentricity of the laminations. The wear of all moving parts of the die becomes uneven and more rapid. In spite of this the machine gives satisfactory results in practice.

Most of these disadvantages can be avoided if a tongue-type feed arrangement is introduced between individual operations, the progress of work being shown diagrammatically in Fig. 2. The adoption of this feed permits the use of single-operation dies of small dimensions, while the results obtained compare favourably with those of follow-on presses. This type of feed has the disadvantage of relatively slow speed due to the number of motions of the tongue mechanisms. Press tools equipped with it operate satisfactorily at a speed of 60 to 80 operations per minute. A 150-ton press produces stator laminations of 250 mm diameter at a rate of 60 per minute. This design shows high rigidity and very little recoil.

A press of this type will produce 24,000 stator and as many rotor laminations per shift of 8 hours, which is the number required for 120 motors of 5 kW rating. The above figure takes into account the setting-up and grinding-in of dies and normal interruptions of work.

The output of the machine can be further increased

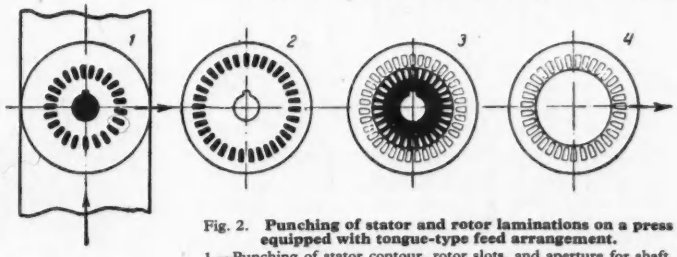


Fig. 2. **Punching of stator and rotor laminations on a press equipped with tongue-type feed arrangement.**

1.—Punching of stator contour, rotor slots, and aperture for shaft.
2.—Punching of stator slots.
3.—Punching and stacking rotor laminations.
4.—Stacking of stator laminations.

if a mechanism is added for automatic stacking of laminations on top of each other. As soon as the required number of laminations for one stator is obtained, these are removed automatically and a new stack is commenced.

The three improved methods of punching, i.e., follow-on dies, twin horizontal compound dies and press tools using tongue feed show no marked superiority with respect to each other. The output obtained by each of the three types is approximately ten times higher than that obtained by the compound presses used at present.

Jig for Turning Cam Profiles in a Centre Lathe

By J. MÜLLER. (From *Werkstatt und Betrieb, Germany*, Vol. 81, No. 7, July, 1948, p. 185, 3 illustrations.)

A PORTAL-TYPE frame (*d* in Fig. 3) supports, in fixed bearings, an upper spindle driven by pinion *a*, Fig. 1, on the lathe headstock spindle. A cradle (*c*, Fig. 3), pivoted on the upper spindle carries the swinging work spindle, on which is mounted the work-piece *d*, Fig. 1. The cradle radius is chosen fairly large so that the maximum vertical displacement of the work spindle remains small, even for large cam radii, i.e., large lateral displacements of the work spindle. Profile distortions due to vertical spindle displacement against the tool tip are thus minimized. A master cam (*c*, Figs. 1 and 2) on the work spindle next to the work-piece is pressed against roll *d*, Fig. 2, by means of a weight *g*, Figs. 2 and 3. As the spindle and master cam turn, the work spindle oscillates towards, and away from, the cutting tool in the top slide. The work-piece is turned from the side, with axial feed, so that the tool position need not be changed for correct angular positioning, and only the cutting width alters as the spindle swings in the course of one revolution. Pronounced concave cam forms can only be turned under certain conditions, but convex forms can always be easily cut.

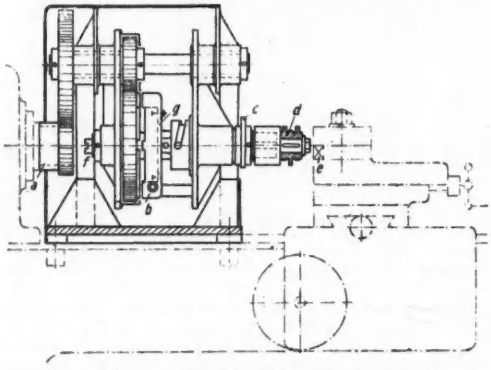


Fig. 1. **Axial section through profile turning jig.**
a = driving pinion on headstock spindle, *b* = dividing disk with fine adjustment, *c* = master cam on swinging work spindle, *d* = blank for several cams, *e* = side-cutting turning tool on lathe top slide, *f* = positioning slot for cam template, *g* = graduated dial.

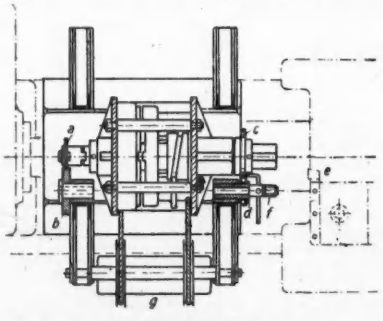


Fig. 2. **Plan view.**
a = Original cam template, *b* = guide roll, *c* = master cam, *d* = roller for master cam (to be removed when marking out master cam), *e* = side-cutting turning tool, *f* = pin with marking-out needle, *g* = weight pressing master cam against roller.

The front end of the work spindle carries a key which positions the work blank *d*, Fig. 1. A slot *f*, Fig. 1, at the rear end of the work spindle coincides with the position of the front key and, in combination with an inside taper, serves to take a chuck on which is carried the original cam (taken from a working model) or a cam template made of steel sheet (*a*, Fig. 2). This cam template only serves to mark out the master cam. The template is inserted in the work spindle and pressed against guide roll *b*, Fig. 2, whilst the spindle is turning. The roll guiding the master cam is removed and a blank for the master cam fixed on the spindle. A marking-out needle is then pushed through a hole in pin *f*, Fig. 2, adjusted to the radius of the master roll just removed, and the master blank is marked out by the needle when the spindle is slowly turned and swings according to the template profile. Several cams can be cut on one blank and their relative angular position assured by arrangement of a dividing disk with corresponding grooves and keys (*b*, Fig. 1). Fine adjustment is provided by a worm gear and the correct position can be read from a graduated dial (*g*, Fig. 1).

All gears are enclosed, and the whole jig is of fabricated construction. The weight of concrete is held by ropes running over wheels. The whole jig is clamped on to the lathe bed by means of specially adapted clamps. The work spindle speed is lower than the headstock spindle speed because of the arrangement of two pairs of spur gears, but the range of speeds is satisfactory for profile turning. Since the jig is mounted as a whole unit, the setting-up times are relatively short, and machining times are also quite normal. The jig has been used in conjunction with a 10-inch centre lathe of 5-ft distance between centres, the jig itself occupying about 2-ft 4-in of this distance. In combination with a normal centre lathe, the jig constitutes a satisfactory substitute for a special profile turning lathe.

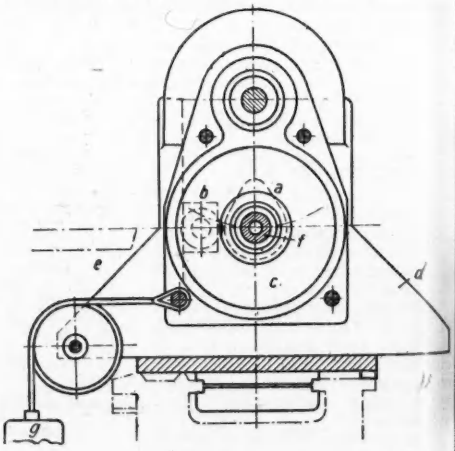
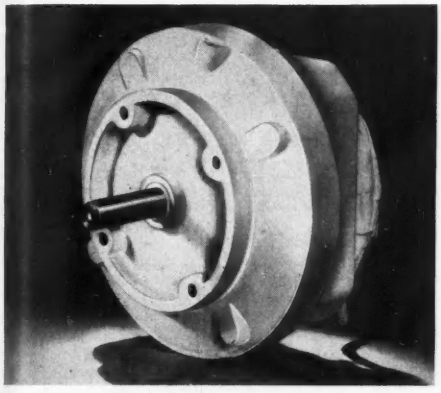


Fig. 3. **Radial section viewed from headstock end.**
a = cam, *b* = roller, *c* = cradle, *d* = portal-type frame, *e* = turning tool shown out of position, *f* = swinging work spindle, *g* = weight.

New Materials, Processes and Equipment

NEW SHADED POLE SINGLE BEARING MOTOR

Designed as a single bearing motor to ensure accurate line-up in the air gap, this new unit is of the four-pole type with a no-load speed of 1700 rpm and a full-load speed of 1550 rpm. The single bearing actually consists of two sleeve bearings, both self-lubricating, with ample space between them to provide an oil reservoir which is packed with felt.



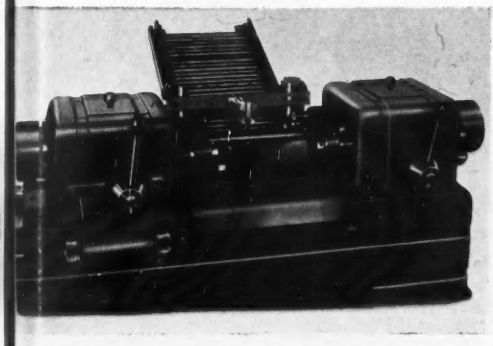
This feature, together with the accurately ground rotor and precision bored stator, provide the quiet running necessary for applications such as wire recorders, turn tables, fans, etc.

The motor shown is of open type design, but can also be supplied with cover to make it totally enclosed, or with cover having a back opening for use with internal cooling fan. Three models are available with continuous duty ratings of 1/100, 1/80 and 1/60 hp. Larger ratings can be furnished for intermittent service.

The motor is designed and manufactured by Electro Engineering Products Co., Inc., Chicago 44, Ill.

AUTOMATIC CENTERING MACHINE

The illustration below shows a close-up of the rear of the Automatic Centering Machine manufactured



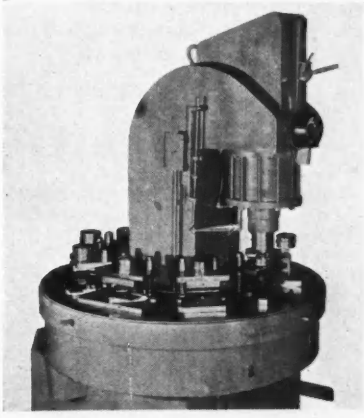
by the Manchester Repetition Engineers Ltd., England. The components are loaded into a gravity fed magazine and are selected by two vees which lift the components to an approximate centre height.

Two headstocks are mounted on the bed of the machine, one of which has an independent slide; this headstock is attached to a cam operated bar having a 2-in travel. Both heads are provided with spring loaded arms which carry hardened and ground cones. These cones take hold of each end of the workpiece during the forward traversing motion of the head. A dwell on the cam causes a hesitation of the head whilst the vees return to the magazine. Continuation of the traverse centralizes, secures, and centres both ends of the workpiece. Independent stops control the depth of each centre hole.

The return stroke of the head allows the workpiece to fall upon a separating plate and is finally ejected into the trough of the machine.

HYDRAULIC TURRET PRESS

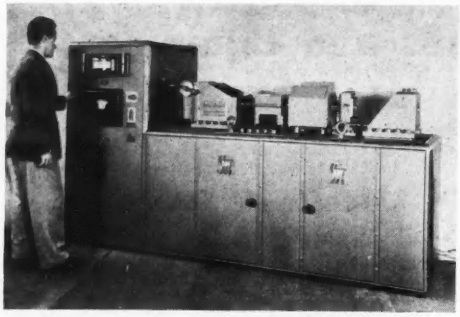
A New Hydraulic Turret Press has been announced by the Rettig Engineering Company to be distributed through the Universal Airline-Joint Mfg. Co., Lafayette, Indiana. This press was designed to meet the demand for a fast press that would practically eliminate tear down and set-up time necessary with conventional type presses. It is especially adaptable on short runs where a schedule per day, week, or month is required.



This press has a large Turret Table that will accommodate as many set-ups (depending on their size) as will fill the table. After initial mounting of fixtures, the operator in a minute's time can change from one fixture to another whether the operation is blanking, forming, drawing, swedging, assembling, etc. These hydraulic turret presses are built in 3 sizes, 10, 20, and 30 ton and may be operated either automatically or manually.

TURBINE BLADE FATIGUE TESTER

A new fatigue testing machine, designed to subject turbine blades and materials to fatigue loads under conditions similar to those in high temperature turbine service, is announced by Baldwin Locomotive Works, Philadelphia. This machine applies alternating flexure loads up to ± 1350 lb to a specimen at a frequency of 3600 cycles per minute while it is held at temperatures



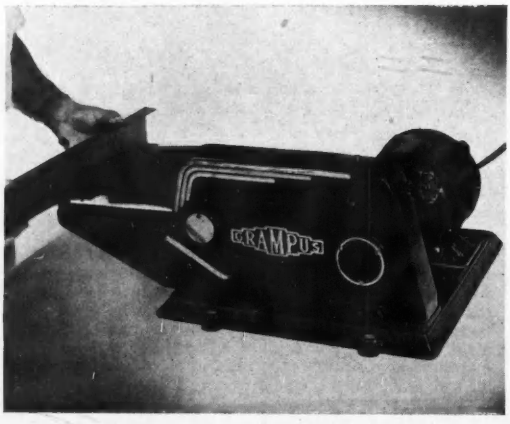
up to 1800° F and is under tensile loads up to 8000 lb. Test specimens in the furnace are held on each side by grips in a fixed head and an oscillating head, as illustrated.

Flexure loads are applied by the centrifugal force of a mass rotating at constant speed in an oscillating frame. This force is accurately controlled by varying the distance between the mass and its centre of rotation. Inertia forces of this mass are compensated by carefully designed springs and calibrating weights.

Automatic controls maintain a constant tension on the specimen, stop the main motor if the specimen breaks, if creep is greater than $\frac{1}{4}$ inch, and if the amplitude of the oscillator exceeds the setting and endangers flex plate guides; and they maintain furnace temperature and rate of power input.

HIGH-SPEED SHEAR

A new, high-speed, low cost shear called the "Grampus," designed with an unusually small head and base to permit wide variation in work shape, is announced by its national distributors, Federal Machinery Co., 134, Grand Street, New York. Material of almost any odd shape can be easily placed over the head or under the base for quick efficient and burr-free cutting.



The "Grampus" is excellent for fast inside cutting, both straight and contour, by means of a simple locking device. No starting hole is required for inside work. Rated up to and including 14 gauge mild steel, there is finger-tip control of the stroke, and rapid adjustment of the cutting blades for very light material. It will handle pipe work, pans, channels, tubes, shells, etc., of various diameters and depths, within the limits of its 7-in throat.

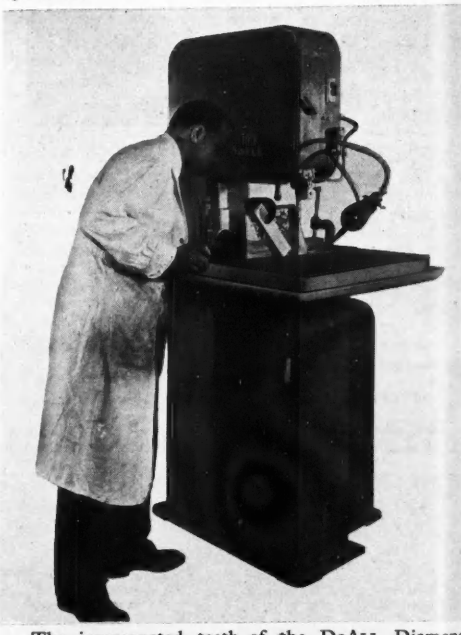
NEW NITROGEN GENERATOR

Gas Atmospheres, Inc., Cleveland, Ohio, announces development of a new low-cost nitrogen generator known as the "Hi-Nitrogen Generator" for use in the metallurgical, chemical, food packaging, refining, processing and storage industries. The self-contained Hi-Nitrogen Generator is a combustion method unit for producing nitrogen at a cost of a few cents per 1000 cubic feet of dry gas. The equipment operates on a mixture of air and natural gas or any manufactured gas such as butane, propane, water gas or coke oven gas.

DIAMOND BAND SAWING MACHINE

With this new machine, developed by the DoALL Company, Des Plaines, Illinois, hardened steel, tungsten carbide, stone and vitreous materials can be cut directly to a layout line with a precision comparable to that of conventional contour sawing. In cutting direct to the desired contour, the material removed is in a salvageable form. It eliminates the conventional method of shaping the material to the desired form by multiple cuts with a diamond impregnated wheel, or successive rough and finish contour grinding operations.

A smooth cutting edge gives close dimensional control and fine finish and there is no vibration or fluttering which frequently shatters friable material when cutting thin work sections. By stacking thin materials and clamping, duplicate parts can be cut in one sawing operation.



The impregnated teeth of the DoALL Diamond Saw Bands are cylindrical segments consisting of diamonds tightly bonded in a sintered tungsten alloy matrix. These segments are electronically brazed to each "tooth" on the steel band. The Sawing Machine has a 16 inch throat depth and 10 inch work thickness capacity. It provides a rigid, smooth 24 inch square box-type work table, tiltable 45 deg. to the right, 10 deg. to the left, front and rear and fitted with splash guards. A coolant pump, coolant reservoir and drain troughs are provided. The machine is also equipped with one set of special saw guides and blades wipers.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

ELECTRIC INSTRUMENTS

Voltage Indicator for High-Voltage Installations

By V. SELMECI. (From *Elektrotechnika, Hungary*, Vol. 40, No. 10, October, 1948, pp. 224-225, 1 illustration.)

LOW-PRESSURE gas-filled discharge tubes are usually employed to determine whether a certain part of an installation is alive. The tube should light up brightly in the electric field of parts under voltage. These tubes have two metallic electrodes, which in the course of time absorb the gases, so that, in consequence, the tube becomes useless. To make certain that the indicator is reliable, it is tested before use by bringing an object under lower voltage into contact with one of its electrodes.

This obvious disadvantage in the use of the indicator has been overcome by the introduction of the Burger-type voltage indicator. This consists of three parallel gas-discharge tubes. The two outer tubes have no metallic electrodes and their working life is theoretically unlimited. The central tube does not light up and serves only for comparison. By observing the central tube it can be determined whether the outer tubes are lit up by the presence of an electric field or only reflect some external light source.

Tests on the 10-kV type indicator showed glow from 5 kV upwards. When the indicator was lit by a 300-W lamp from a distance of 5 ft, the glow was easily discernible. Using the indicator on a 100-kV transmission line, the indication was visible from a distance of 200 ft. The current passing through the indicator was 1 mA.

The fitting of these tubes should be advantageous where the use of instruments is not warranted and only a constant indication of the voltage state is required.

HEAT TREATMENT

Surface Hardening Methods

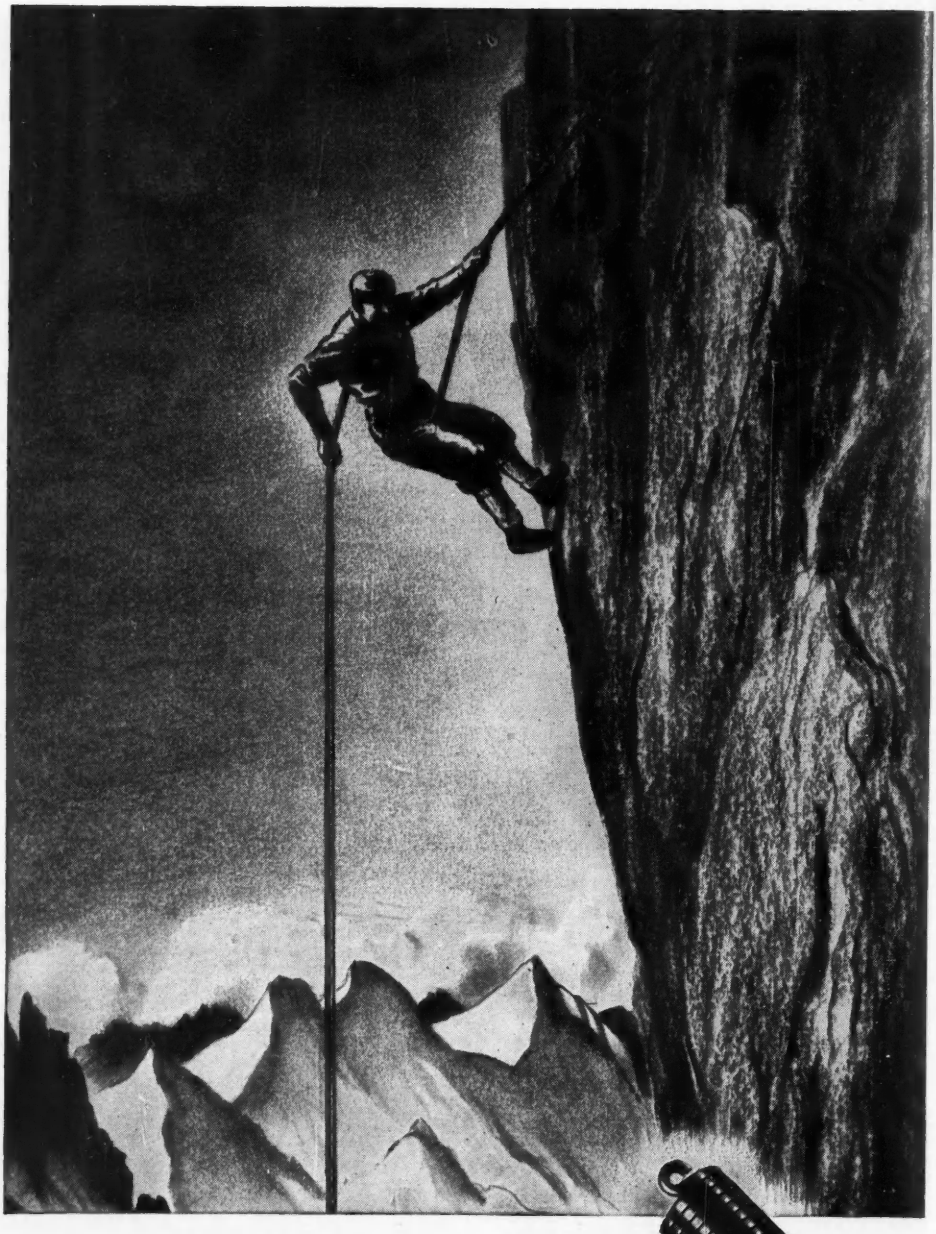
By S. FÖRSTER. (From *Werkstatt und Betrieb, Germany*, Vol. 81, No. 6, June, 1948, pp. 151-157, 5 illustrations.)

Flame Hardening of Gears

By W. GRÖNEGRESS. (From the same journal, pp. 145-150, 17 illustrations.)

THE first article surveys the principles and different methods of, as well as suitable steels for, the various known surface hardening processes, i.e., case-hardening, nitriding, differential critical cooling rate of special steels, dip hardening in hot melt baths, gas flame hardening and electric induction hardening. Data regarding the attainable hardening depths are given. The main part of this article deals with equipment for induction hardening as has been evolved in Germany during the war. A great variety of steels can be hardened, amongst them high-carbon steels, tool steels, and high-speed steels (after previous homogenizing treatment). Finished tools, such as drills, chisels, reamers, etc., can also be conveniently hardened. Main advantages of the induction hardening methods are their adaptability to the size, shape and material of the work, the short time taken and the heat concentration, which avoids the danger of cracking even with high-carbon steels (e.g., 0.60 per cent carbon). It is the cheapest way of hardening specially valuable steels where required and it often enables the use of cheaper steels in substitution. The equipment consists of precision apparatus and only series or mass production is really economical, except where special steels would, by any other method, require extensive treatment. The frequency of the electric current depends on the shape and size of the work and the required depth of hardening. Large crankshafts have been hardened, to a depth of 2 inches, with mains frequency. Medium frequencies ranging up to some 10 kilocycles can be supplied by rotary generators, but the plant is cumbersome and requires a great amount of energy. 50 kVA was required at the generator to harden a milling cutter of $2\frac{3}{4}$ inches diameter, and $\frac{3}{4}$ inch thickness, a frequency of 30 kilocycles producing a case depth of 0·1 inch. A 12-kW generator is used for hardening cutting tool edges. High frequencies are produced by valve generators fed from 12-kV rectifiers which can be connected to 380 Volt a.c. mains. High-frequency hardening plant experienced rapid development during the war years in Germany, and a series of 5, 20, 40, 100, 200 kW plant was standardized and equipped with ordinary transmitter valves. Several totally enclosed generators and control boxes were arranged side by side in workshops and a greater number of hardening machines placed on rails nearby, so that machines could be connected or disconnected according to the production schedule. The plant is described in detail. 5000 medium size gears could be hardened per month with one of these machines.

Gear teeth above 5 D.P. can be gas flame-hardened tooth by tooth with sieve or slot burners adapted to the tooth form and fed at constant speed across the face width, followed at $\frac{1}{2}$ to $\frac{5}{8}$ inch distance by the water spray. Gas consumption is 0·9 gallon town's gas with 0·6 gallon oxygen per minute for one 0·04 inch sieve hole, or half that amount per 0·04 inch slot width. The optimum gas velocity is 460 fps. For gears of relatively uniform size, an auxiliary hand-controlled jig with a burner carriage may be sufficient, but universal and automatic hardening machines have been developed for the treatment of spurs, bevels, worms, and wormwheels of various sizes, and for mass production, guaranteeing closely controlled hardness figures. Unalloyed steels are used for core strengths of less than 60 tsi, and alloyed steels for higher core strengths. Gears with teeth smaller than 5 D.P. are treated by the rotation process, i.e., they are rotated through the flame, heated to hardening temperature and quenched by spray or in a bath. Although the gear core remains soft, the teeth are fully hardened in this process. Flame hardening has been successfully employed on a large scale. The advantages are the possibility of using cheaper, unalloyed substitute steels, increase in wear and rolling fatigue strength, which makes possible a reduction in gear size, the possibility of hardening very large gears with only a small amount of heating energy, and minor hardening distortions which, according to experience, remain entirely within the permissible machining errors. Danger of cracking is reduced, the time saving is considerable, finish grinding can in most cases be omitted or, if required, the grinding allowance can be considerably reduced. The hardening of wormwheels has become possible by the introduction of this process.



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INSULATING MATERIALS

The Application of Heat-Resistant Insulating Materials to Electric Motors

By M. RIGGENBACH. (From *The Brown Boveri Review, Switzerland*, Vol. 35, Nos. 5/6, May/June, 1948, pp. 143-146, 2 illustrations.)

THE appearance on the market of new heat-resistant insulating materials for electrical machinery allows of designing electric motors to operate reliably at temperatures considerably higher than those permissible in motors insulated with the conventional type of insulating material. Since it has become possible to produce a practical glass-fibre covering which is much less voluminous than asbestos insulation, and which can be impregnated with heat-resistant varnishes, e.g., of the silicone type, operating temperatures of the order of 150° C or more can be allowed without incurring the risk of breakdowns.

Instances where heat-resistant insulation should be employed are described in this article. Some practical examples serve to illustrate the fact that the new heat-resistant insulating materials can be used with advantage for motors which are frequently started or reversed, and also when the space available is limited or the ambient temperature very high. However, for continuously operating standard motors, the use of these new materials is not warranted, since, at present, their cost is too high and, therefore, from an economical point of view, no advantages would accrue, despite the possible increase in output at higher temperatures.

ILLUMINATING ENGINEERING

A Rapid-Action Starter Switch for Fluorescent Lamps

By TH. HEHENKAMP. (From *Philips Technical Review, Holland*, Vol. 10, No. 5, November, 1948, pp. 141-149, 15 illustrations.)

WITH the usual types of starter switches for fluorescent lamps, namely, resistance and glow-discharge bimetallic starters, several seconds elapse between the switching-on and the ignition of the lamp. Though in many cases of public utility lighting this is no drawback, it is of more serious consequence in domestic lighting, in which field the fluorescent lamp is gaining more and more ground. To overcome this difficulty, a new solution has, therefore, been sought, based on recognition of the fact that the cathodes should first be sufficiently heated so as not to suffer any damage when the lamp is ignited. For this heating to take place in a reasonably short time (0·3 to 0·4 sec.), a current has to be sent through the filaments which is stronger than the stationary short-circuit current of the series choke. This heating current is obtained by utilizing a transient, the circuit being periodically closed and opened by a contact forced to vibrate by means of a coil shunted across it. The time normally elapsing between the switching-on and the ignition of the lamp is 0·3 to 0·4 sec. and, in unfavourable circumstances, 0·5 to 1 sec. The new starter switch is interchangeable with the conventional glow-discharge bimetallic starter.

LUBRICANTS AND LUBRICATION

Observations of the Lubricating Oil Film Between Piston Ring and Cylinder of a Running Engine

By J. C. WISDOM and R. L. BROOKS. (From *Report ACA-37* of the *Australian Council for Aeronautics*, July, 1947, 28 pages, 19 illustrations.)

A METHOD of examining the lubrication between piston ring and cylinder by measurement of the electrical contact resistance has been applied to a single-cylinder aircraft engine running under its own power up to speeds of 2000 rpm. The effect of engine-operating conditions on ring lubrication has been observed and it is concluded that, with the ring combination used, full fluid lubrication never exists for the whole of the cycle. The breakdown of the lubricating oil film at the upper position of piston travel has been observed, and it has been demonstrated that factors which facilitate breakdown are low engine speed, high gas pressure in the cylinder, low oil circulation, and high cylinder temperature. This is in agreement with known data on cylinder wear. Ignition advance beyond the optimum point causes increased breakdown, and, if detonating conditions obtain, greatly increased wear near the top of the cylinder would take place.

MACHINE SHOP PRACTICE

Grinding of Hard Alloys

By H. KALPERS. (From *Metalloberfläche, Germany*, Vol. 2, No. 8, August, 1948, pp. 171-173, 2 illustrations.)

HARD alloys are sintered carbides of tungsten, titanium and other metals, with cobalt and other metallic additions. Their hardness is extremely high and cannot be changed by any heat treatment, so that grinding for cutting use must be done very carefully. Hard metal tips are usually furnace-brazed onto a 0·5 to 0·7 per cent carbon-steel shaft by means of very thin foils of electrolytic-copper wire mesh which prevent stresses due to differential contraction of the shaft and tip. Heating and brazing are done in a reducing atmosphere and the brazed tool is very slowly cooled to room temperature before grinding commences. At first, the clearance angles and support face of the shaft are ground with a normal corundum disk. The clearance angles of the shaft should be chosen about 2 degrees larger than those of the hard-alloy tip. The tip is ground with silicon carbide disks of definite grain size and hardness, which are chosen specially for pre-grinding, finish-grinding or precision-grinding. First the top face is ground, this being followed by grinding of the clearance faces. Care must be taken that the silicon carbide disk is not smeared over by the accidental grinding of parts of the softer shaft material. The disks must not be too hard, as the hard-alloy tip may crack under too hard a disk. Grain size may be fairly coarse for pre-grinding, but should be fine for finish-grinding and especially fine for precision-grinding. Since precision-grinding increases the tool life, it is recommended for all finishing tools. When re-grinding is necessary, it is more economical to re-grind before the tip is blunted and the cutting efficiency decreased. If done in good time, re-grinding is cheaper than leaving the tool until it becomes blunt. The steel should always be allowed to cool down completely before re-grinding is started. Grinding by hand should be done at a grinding speed of 80 fps, and automatic grinding with special silicon carbide disks at a grinding speed of 20 to 50 fps. Grinding pressure must not be too great, as otherwise cracks are formed and the disks are worn down without any improvement to the ground faces. The disks should be worn down uniformly over their whole width and their diameter must not be too small, since concave clearance faces reduce the support of the cutting edge. Dry grinding must be done with very great care so that the temperature distribution in the tip remains fairly uniform. The tip must not be cooled in water after grinding, since any rapid cooling would lead to cracks. Suction fans for the grinding dust are required. Wet grinding requires a regular ample flow of clear water of about 68 to 77° F and cooling must be uniform over the whole tip. For cutting materials giving long chips, a special chip-guiding ledge should be ground adjacent to the cutting edge, using a flat disk of very fine grain and medium hardness, about 6 inches in diameter and $\frac{3}{4}$ inch wide, formed to a slight cone in order to obtain a very sharp cutting edge for the tip.

TESTAL

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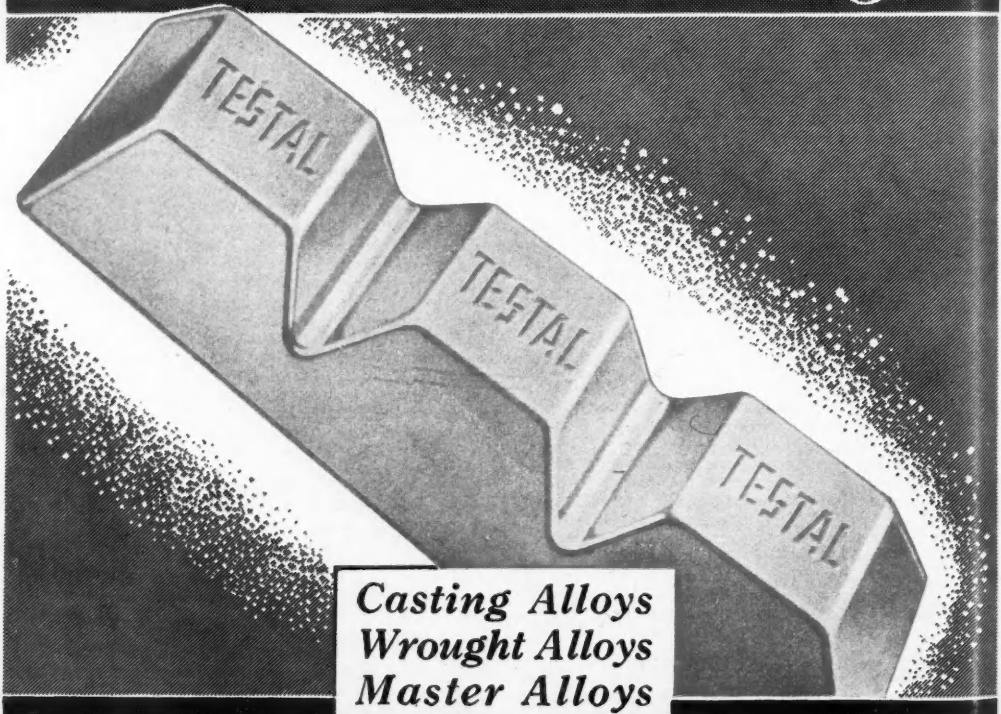

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NEWS OF THE MONTH

PERSONAL

Mr. H. Annis has been appointed chief inspector of Metropolitan-Vickers Electrical Co., Ltd., Trafford Park, Manchester 17.

Mr. George Archer, C.M.G., and **Mr. L. K. Brindley, M.B.E.,** have been appointed directors of the Mond Nickel Co., Ltd., Grosvenor House, Park Lane, London, W.1.

Mr. W. Fillingham Brown, B.Sc. (Eng.) (Lond.), M.I.C.E., has been appointed chief engineer and general manager of the Colne Valley Sewerage Board, 20 Castle Street, Hertford, Herts.

Mr. H. A. Chambers has been appointed general manager of the Rockwell Machine Tool Co., Ltd., Second Way, Exhibition Grounds, Wembley, Middx.

Mr. W. O. Fenwick, M.I.E.E., has been appointed technical editor of the Electrical Review.

Sir Arthur P. M. Fleming, C.B.E., D.Eng., director of research and education of the Metropolitan-Vickers Electrical Co., Ltd., Trafford Park, Manchester 17, has been elected President of the Engineering Section of the British Association for the year 1949, when the meeting will be held in Newcastle-on-Tyne.

Mr. A. G. Galle, A.M.I.E.E., M.Inst.W., has been appointed Midland technical sales representative of Sciaky Electric Welding Machines Ltd., at the firm's Birmingham office at 13 Navigation Street, Birmingham 2.

Mr. J. C. Gridley, executive director of Powell Duffryn Ltd., will be appointed as from January 1, 1949, Chairman of the Vacuum Oil Co., Ltd., Caxton House, London, S.W.1. **Mr. H. W. Rocke** will continue in his position as managing director of the Company.

Mr. M. R. Haddock, O.B.E., has been appointed assistant secretary of the Docks and Inland Waterways Executive, 22 Dorset Square, London, N.W.1.

Mr. C. P. Harrison has been appointed manager of the Marine Department of the General Electric Company, Ltd., Magnet House, Kingsway, London, W.C.2.

Mr. A. P. MacKim, A.M.I.E.E., has been elected technical engineer to the South West Scotland Electricity Board, 62 Dalziel Drive, Glasgow, S.1. **Mr. J. S. Helton** has been appointed purchasing officer to the Board.

Mr. R. E. Marks has been appointed manager of the Government Railway and Overseas Department of the Vacuum Oil Co., Ltd., Caxton House, London, S.W.1.

Mr. E. A. Milne has been appointed to take charge of the carriage and wagon department of the Railway Executive, Scottish Region, at Cowlairs works, Glasgow, N. **M. J. S. Scott** has been appointed works manager of the locomotive shops, and **Mr. E. R. Parsons,** works manager of the carriage and wagon shops at the Cowlairs works. **Mr. J. Bull** has been appointed works' manager of the locomotive shops and **Mr. H. Wilcock,** works' manager of the carriage and wagon shops at the St. Rollox works.

Mr. L. F. Schneider, A.M.I.Ec.E., has commenced an Industrial Consultant's Practice at 12 Elm Way, London, N.W.10. Telephone: GLADstone 6564.

Mr. D. E. J. Talbot, B.Sc. (Wales), has been appointed research metallurgist in the Armaments Research Establishment, Royal Arsenal, Woolwich, London, S.E.18.

Mr. B. J. Tams, M.Sc., M.I.Mech.E., has been appointed general manager of J. Brockhouse & Co., Ltd., Elms Works, Penn Road, Wolverhampton.

Mr. G. W. Tookey has been elected chairman of the Junior Institution of Engineers.

Major-General W. S. Tope, C.B., C.B.E., M.I.Mech.E., Director of Mechanical Engineering, War Office, has been appointed Honorary member representing the War Office at the Council of the Institute of Metals.

Mr. G. M. Wright, B.Eng., M.I.E.E., chief engineer of Marconi's Wireless Telegraph Co., Ltd., Marconi House, Chelmsford, Essex, has been appointed a member of the Radio Research Board of the Department of Scientific and Industrial Research.

BUSINESS NOTES

Air Control Installations Ltd., of Ruislip, Middlesex, announce the opening of a branch office in Birmingham. Mr. A. W. Evans, A.M.I.H.V.E., who has many years' experience of all aspects of air treatment and fan engineering, has been appointed Midland Area Manager. Enquiries should for the present be addressed to Mr. Evans at 6 Buryfield Road, Solihull, Birmingham. Telephone: SOLihull 1718.

Associated Lead Manufacturers Ltd., Ibex House, Minories, London, E.C.3, has been formed to incorporate as from January 1, 1949, the businesses of A. T. Becks & Co., Ltd., The Cookson Lead & Antimony Co., Ltd., The Librex Lead Co., Ltd., Locke, Lancaster & W. W. and R. Johnson & Sons, Ltd., The London Lead Oxide Co., Ltd., The Oidas Metals Co., Ltd., and Walkers, Parker & Co., Ltd.

Bristol's Instrument Co., Ltd., has transferred its head office to a new factory at Weymouth, Dorset. Part of the firm's premises at Brent Crescent, North Circular Road, London, N.W.10, will be maintained as a London office.

Cox & Danks Ltd. has removed its South Wales branch to Scapa Works, Neath, Glam. Mr. W. R. Jones is Branch Manager.

E. H. Jones (Machine Tools) Ltd., Edgware Road, The Hyde, London, N.W.9, has appointed **Bond & Partners,** 25, Collingwood Street, Newcastle-on-Tyne 1, representatives for the North Eastern area.

A. A. Jones & Shipman Ltd., Narborough Road South, Leicester, have removed their London office to Murray House, 5 Vandon Street, Buckingham Gate, London, S.W.1. Tel. WHITEhall 0508.

Philip & Son Ltd., Shipbuilders, Sandquay and Noss Engineering Works, Dartmouth, removed their London office to 131 Victoria Street, London, S.W.1.

Tube Investments Ltd., have formed a new company, the TI Aluminium Ltd., to co-ordinate the administrative, technical, research and marketing resources of the group's aluminium division. The division includes Reynolds Light Alloys Ltd., Reynolds Rolling Mills Ltd., and the South Wales Aluminium Co., Ltd. The head offices of the new Company will be at Redfern Road, Tyseley, Birmingham.

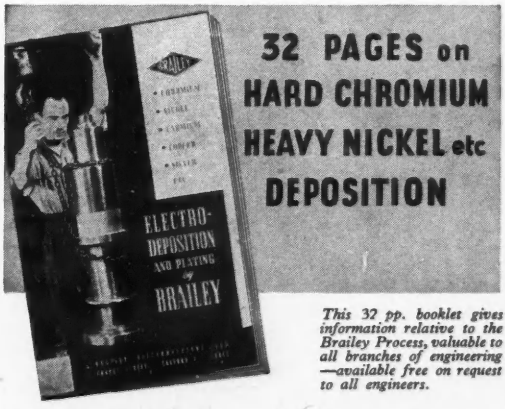
Scophony Ltd. has acquired the assets and undertakings of John Logie Baird Ltd., television manufacturers, and its associated company, W. Andrew Bryce & Co., Ltd. Mr. J. Buchanan, managing director of the two companies, has joined the board of Scophony Ltd.

ENGINEERING AND THE PROPOSED IRON AND STEEL BILL

THE Engineering Industries Association representing the views of 3,500 British engineering concerns has issued a statement declaring itself fundamentally opposed to the proposed Iron and Steel Nationalisation Bill. The statement reads:—

"This Association is deeply concerned with the effect the proposed Iron and Steel Nationalisation Bill will have upon the Engineering Industry of Great Britain. This Association, composed mainly of steel consumers, view the proposed measure as one which will inevitably have a disastrous effect upon their great industry.

"Its immediate effect will be to create an atmosphere of uncertainty in which forward planning—an integral part of engineering—becomes almost impossible; an atmosphere which inevitably will last for more than a year and a half.



This 32 pp. booklet gives information relative to the Brailey Process, valuable to all branches of engineering —available free on request to all engineers.

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" It is to be deplored that such a state of affairs should be brought about at a time when our very existence as a Nation is dependent upon the productive efficiency of the Engineering Industry.

" Many of our members, under the provision of the Bill, will have to buy their raw materials from state owned concerns. How can they view with equanimity a position in which their supplies of raw materials will be controlled by their competitors ? If this Bill reaches the Statute book private firms will have to compete against those firms owned by the Iron and Steel Corporation and backed by the whole weight of the Government. Will the Government be likely to give a firm operating under private enterprise the same facilities as their own—surely there exists the danger of preferential treatment.

" The Steel Industry has long been outstanding in its productive capacity, its technical skill and its labour relations, due to good management. We have already seen the gradual decrease of managerial authority in the Nationalised coal industry. Is this experience to be repeated ? From the moment that Directors no longer have the final word—it is almost inevitable."

Captain C. A. Kershaw, Secretary General of the Engineering Industries Association commenting on the statement said :—

" Our members are mainly steel consumers, therefore, we are intensely interested in the proposed provision for Consumer Committees to protect the users of steel. If this Bill becomes law it is essential that the engineering industries will have full representation in this connection—and by full is meant the interests of both the small and the large consumer.

" It is a matter of the greatest importance to know whether or not the Iron and Steel Corporation means to confine itself to the basic processes and whether it will dispose of those subsidiary companies engaged in activities not producing ore, not smelting, not dealing with ingots, and not working on hot rolling.

" In other words, will this nationalised organisation compete with private enterprise in, for example, the field of engineering and in other enterprises far removed from the manufacture of steel.

" The present excellent relations in the Industry are also threatened by this Bill. The State is to become the main employer, but not the sole employer, nor, in some cases, even the largest employer. What happens to collective bargaining ? "

ANGLO-AMERICAN-CANADIAN UNIFICATION OF SCREW THREADS

THE need has long been felt for a standard screw thread series which could be used when it was desirable that products should be interchangeable in whatever country they were produced. As long as 40 years ago, discussions were commenced between the British Standards Institution and its counterparts in the U.S.A. In 1943, 1944 and 1945 a series of conferences took place between the United Kingdom, the U.S.A., and Canada, and agreement has now been reached between the three countries on the thread form and a series of diameters and pitches for bolts, nuts, and fastening threads for sizes ¼ in. and above.

The three countries primarily concerned with the details are now preparing full particulars of the agreed standard and it is hoped that the other Commonwealth countries will be able to adopt the same standard.

The thread system is to be known in all three countries as the Unified Screw Thread system. The form of thread is one having a 60 deg. angle. The bolt thread will have a rounded root with the alternatives of a rounded crest for use in the United Kingdom and flat crest for use in the U.S.A. The pitch diameter series will comprise a fine and a coarse series, and provision will be made in the standard for special design threads.

For the sizes below ¼ in. the proposals put forward by the U.S.A. provide for two series which have pitches finer and coarser, respectively, than the British B.A. series. From an engineering point of view it is difficult to believe that there is a need for two series so close together and the subject is still being discussed in the United Kingdom. In the meantime, sizes smaller than ¼ in. will not be included in the Unified system.

The issue of the standards for the Unified Thread System in the three countries does not imply that there must be an immediate change-over by British industry to the new screw thread. Obviously the adoption of the new system will be advantageous where servicing requirements and replacements call for interchangeability in products made in any one of the three countries, but it will be the responsibility of each section of British Industry to decide for itself whether it wishes to adopt the new system.

This agreement, and the adoption of the new system in the three participating countries, and possibly also by the other Commonwealth countries, must, however, be regarded as a most important step in practical co-operation between the nations, and one which must inevitably save each country vast expenditures in time and money.

BROCHURES AND CATALOGUES

Fully Automatic Die Casting Machine. The Madison-Kipp No. 4 fully automatic die casting machine for the production of zinc, lead and tin pressure die castings is now manufactured in England by Wm. Coulthard & Co., Ltd., Crown Works, Carlisle. An illustrated folder just issued contains a rather comprehensive description of the interesting machine with complete specifications. Net weight of the machine is 14,000 lbs., the size (distance between guidebar centres) 12 in. × 16 in. Standard equipment includes 3 h.p. variable speed, constant torque A.C. motor, Electric control drum, Air pressure gooseneck and Gas burner and blower. Operation of the machine is extremely simple with only one operating lever. Copies of the folder and further information is gladly supplied on application to the makers, Wm. Coulthard & Co., Ltd., Crown Works, Carlisle.

Oxygen—Its Potentialities in Iron & Steel Production. The British Oxygen Co., Ltd., of Grosvenor House, Park Lane, London, W.1, has published an extremely interesting 60-page booklet on Oxygen and its application to metallurgical processes. The booklet provides an excellent summary of the work accomplished in Great Britain, America, Germany and Russia, in connection with the use of oxygen in iron and steel production. Experimental work is at present being carried out between the steel industry and the oxygen industry through the agency of the British Iron & Steel Research Association, and the British Oxygen Company has completed all technical preparations for the building of medium purity oxygen plants.

G.E.C. Cold Pressure Welding Aluminium. In view of the great interest aroused by the recent announcement of this process it has been decided to publish an illustrated booklet giving further details of the method of welding metals at room temperature. Although the process may be successfully applied to several types of metals, its most important application is for the cold welding of aluminium. Copies of the illustrated booklet may be obtained from the General Electric Co., Ltd., Magnet House, Kingsway, London, W.C.2.

Terra-Tertia Test Unit. Electro Methods Ltd., of 112 Brent Street, London, N.W.4. has published a leaflet with detailed description of the Terra-Tertia Test Unit which provides a simple means of maintaining electrical equipment in a safe and foolproof condition. Danger is indicated by the failure of the lamp—a unique feature of the unit which can be handled by unskilled persons. The unit is easily installed. Further details will be supplied on request by Electro Methods Ltd., 112 Brent Street, London, N.W.4.

BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 24 Victoria Street, London, S.W.1.)

British Standards for the Automobile Industry. (Handbook No. 8). This Handbook brings together in readily accessible form the standards of direct interest to the automobile industry and all concerned with it. Its purpose is to facilitate the work of people who have frequent need of standards, such as production engineers, design engineers, technicians, draughtsmen, foremen, storekeepers, fitters and many others.

In abridged form, the contents are :—

13 automobile standards of the Society of Motor Manufacturers and Traders (which have now been adopted as British Standards).
19 provisional automobile standards of the Society of Motor Manufacturers and Traders.
10 British Standards, in full, or those parts which are of direct interest to the automobile industry.
References to 18 British Standards of general interest.
Horse power correction charts.
S.A.E. standards for splines and serrations (reproduced from the S.A.E. Handbook).
Steels for the motor industry in the form of bars and forgings ; sheet and strip ; wire for springs.
References to British Standards for aluminium and aluminium alloys, bronzes and brasses.

The Handbook (335 pages) will be edited every two to three years for the inclusion of additional standards and improvement of existing ones. The present edition is priced at 15/-.

Endless V.-Belt Drives (B.S. No. 1440 : 1948). This new standard deals with endless V-belts having an included angle of 40° and consisting of rubber and textile materials ; and with V-groove pulleys, used primarily for power transmission for industrial purposes. It does not cover automotive V-belt drives and variable speed drives employing pulleys with movable flanges.

In one section it deals with belts of M, A, B, C, D and E cross sections, the nominal inside lengths of belt covered ranging from 15 to 358 inches dependent on the cross section. It specifies nominal cross-sectional dimensions and pitch lengths with the permissible variations and defines how these lengths shall be measured.

A further section deals with the standard dimensions of V-grooved pulleys and sets out recommendations in regard to standard pitch diameters, installation and take-up allowances.

An Appendix gives recommended practice for the power application of V-belts and includes power correction factors for different types of service, arc of contact and horse power ratings for the various V-belt sections in relation to the pitch diameter of smaller pulleys. Price 3/-.

High Carbon Oil Hardening Steel (Precision Finish) (B.S. No. 1423 : 1948). The standard was prepared as a result of a request from the clock, watch and instrument industry to meet the need for a precision high carbon steel suitable for machining in Swiss-type automatics.

The chemical composition of the steel and the hardness values are specified, and details are provided of the process of manufacture, finish, microstructure, heat treatment, tolerances allowed and condition of material on delivery. Price 1/-.

Flexible Metallic Tubing for Use in the Petroleum Industry (B.S. No. 1465). The standard was prepared at the request of the Petroleum Industry as there was no specification in existence which could be used by the Industry for their specific applications.

The standard applies to flexible metallic tubing for the conveyance of steam, liquid fuels, lubricating oils and liquid asphalt at various temperatures and pressures. Details of construction and dimensions for various temperatures and pressures are given and a hydraulic test is also provided for. Price 2/-.